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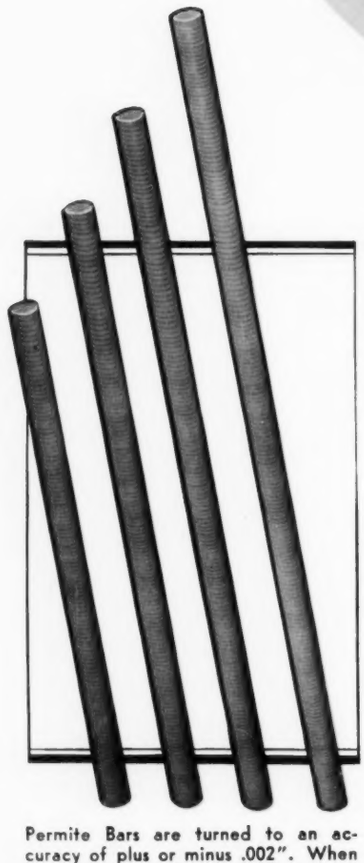

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PRODUCT ENGINEERING

VOLUME 6 NUMBER 3

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Coming Issues

An important detail in all hydraulic systems is the design of the packings used. G. N. Dorr of the Dorr-Patterson Engineering Company has written for the April number a comprehensive article on leather packings, including the design of the gland box, standard packings, and causes of packing failures. This will be followed in the May number with an article by the same author on the design of hydraulic equipment.

The April number will also feature articles on built-in lighting, lubrication, and analysis of design possibilities in the application of hollow-head screws. This last article will emphasize the greater compactness and space-saving that can be attained through the use of hollow-head screws, and will be strikingly illustrated by typical examples.

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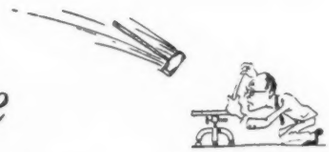
Intimate Correspondence



No Color?

PRODUCT ENGINEERING
NEW YORK

I recently had to dig up some information on color and finishes, which led me to look through some of the early numbers of *Product Engineering*. In issues that dated back to 1932 and earlier I found quite a few articles on color. Some of them had just the information I wanted. But I noticed that for the past year there have been no articles on color. Have you given up printing such articles?

H. E. TAYLOR
Detroit

IT IS true that we have had no major article on color during the past year, although we have had numerous short items on this subject. Also, wherever it applied, we pointed out the part played by color in contributing to the salability of the design. As to the future, we are now negotiating with a recognized authority on the subject for a series of articles on color and finishing materials.

Hopeful Clipper

PRODUCT ENGINEERING
NEW YORK

I take this opportunity again of expressing my appreciation for your technical publication. I belong to your group of clippers.

I have been hoping that you would enter a monthly section on product design such as shown in the April issue 1934.

I am taking the liberty of enclosing under separate cover a problem that has come up frequently.

The enclosed black cap has to be high lighted as you see done in a poor way. We have 300,000 to do this way or any way that would accomplish the same result. The lion cannot be raised due to a trimming operation on the shell.

The powder cover sells for 10c in the Woolworth and there are many such items that are high lighted. At the price they sell for, there must be a cheap method and we are in need of assistance so have solicited your help.

The modern trend has raised this question with us so many times maybe a magazine article dealing on this would be appreciated in the field.

L. E. FABER
Product Engineer
San Francisco

WE ARE not printing Mr. Faber's company connection because we have adopted the policy of omitting such connections on this page, but we have put down his title because it indicates that the function of product engineer is recognized as far west as San Francisco. In replying to Mr. Faber's letter we promised him that the subject he has in mind for an article would be put down on the schedule, and treated just as soon as we found the best way of handling it.

Going! Going!——

PRODUCT ENGINEERING
NEW YORK

Having read with interest the letter of Benjamin S. Malin, in which he asked for reprints of articles on color, I would like to ask for reprints of nearly all of those articles. Like Mr. Malin I became a subscriber to *Product Engineering* too late to have received the articles in their original form. The articles are entitled:

"An Engineer Considers Art in Engineering"—C. K. Lee—May, 1932.
"Color by Trial and Jury"—H. H. Adler—July, 1932.
"What You Can Do With Color"—D. R. Dohner—Jan., 1932.
"What Is Color?"—A. H. Munsell, Maloney, etc.,—Feb., 1932.
"Color—Where and How"—March, 1932.
"Color—Where and How"—April, 1932.
"Color by Optics"—Dr. I. J. Saxl—August, 1932.
"Color—Materials and Behavior in Products"—F. J. Oswald—Sept., 1932.
"Color—Materials and Behavior"—F. J. Oswald—Oct., 1932.

I shall greatly appreciate your courtesy in supplying me with these reprints if they are available.

JOHN A. SHERIDAN
S. Merchantville, N. J.

IF ANYBODY else wants clippings of these color articles he had better speak quickly. Even now there is a question whether we can provide clippings of all of them. At any rate, we have more articles coming, as indicated in the foregoing reply to Mr. Taylor.

Chart Beggar

PRODUCT ENGINEERING
NEW YORK

In regards to the reference sheets and charts that appear in your magazine from time to time, I have a few suggestions that might satisfy all your critics.

I have read, in some of the letters you publish, objection to the size of the sheet. Requests that you publish said sheets on page of even number. One letter mentioned that, as he works for a firm who subscribes to your magazine, the only way he could get this chart or reference sheet for his own was to copy it. I am also in this class, but rarely see the chart as a fellow worker cuts the chart out for his own.

My suggestions follow:

1—Print charts and reference sheets on one side of special paper that will blueprint clearly.

2—Have heavy border the standard note book size for trimming blueprint.

3—This will permanently solve the number of the page as even, for if it was printed on an odd page, a blank sheet would precede the chart and you would hear from some crank in short order about the blank sheet being first.

4—In a drawing room such as I work in all the beggars could get a print of the chart and the firm could file the original.

In my case I know the fellow who confiscates our chart and he couldn't deny me a print of it.

BOTH name and address are off this communication because the writer doesn't want to start a fight with that fellow who swipes the chart. We wish we could adopt his suggestion of arranging charts to be duplicated, but when we tried it we got so many mean comments about the waste of a whole page that we finally gave it up. The answer seems to be a few more subscriptions for that particular engineering department.

From Germany

PRODUCT ENGINEERING
NEW YORK

An inquiry from Germany has been addressed to the writer. The inquirer desires a copy of the articles on cams published in your *Product Engineering* for July, August and September issues and some comments in the January issue. What is the best way of sending him the information? Do you have a special print of the articles in one printing? If not, kindly let us know how much it will cost to have the special issues.

H. E. GOLBER
Chicago

HERE again the easiest answer was to mail clippings. As a matter of fact the issues themselves were not available at any price. We watch our print order carefully and seldom have enough copies left over to take care of more than a very few requests.

In Late

PRODUCT ENGINEERING
NEW YORK

Having recently subscribed, we find that Article 3 and 4 are in the October and December issues on "Retaining and Locking Devices." Please send us issues containing Nos. 1 and 2 of the series, and oblige.

B. H. UHRICH
Independence, Kansas

THAT'S the penalty for not being a charter subscriber, but better late than never. If you come into the family now or later you are almost sure to land in the middle of some important series of articles.

PRODUCT ENGINEERING

KENNETH H. CONDIT, *Editor*

MCGRAW-HILL PUBLISHING CO., INC. MARCH 1935 VOLUME SIX, NUMBER THREE

Objections Over-Ruled

AT VARIOUS times some executives of motor and control manufacturers have expressed objections to publication of informative technical articles about their products. They are willing to foster publicity that extols the virtues of the things they have to sell but they oppose the divulging of data and information that explains principles of operation, service requirements that must be considered, conditions that limit or influence the operation of motors and controls, and similar facts. In most instances the objectors are sales executives.

Those opposed to such informative technical articles contend that even if mechanical designers have but a limited knowledge of motors and controls they will take it upon themselves to select and specify the type to be used, without consulting the electrical manufacturer. This frequently results in serious mistakes or misapplications, and the supplier is put to the task of proving that the electrical equipment furnished meets the ratings, quality and efficiency specified by the mechanical designer. Naturally, such situations, when they do exist, lead to misunderstandings and losses to all parties involved.

Although the contention that ignorance of electrical equipment will compel the mechanical designer to consult the electrical manufacturer may be accepted as true, it does not follow that either a superficial or a good knowledge will deter the mechanical designer from seeking the advice of the electrical manufacturer. The opposite is more likely to be true. As one becomes more acquainted with the peculiarities and intricacies of electrical devices the conclusion is forced home that the correct application of motors and controls requires the aid of specialists.

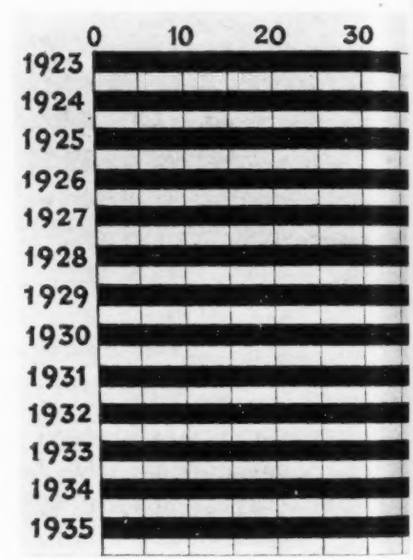
The problem of selecting the right electrical equipment is like an equation containing many variables. One member of the equation sets forth all those factors that determine the demands to be made upon the electrical equipment. This side of the equation must be written by the mechanical designer. The other member of the equation represents the performance ability of the electrical equipment. It is evident that the mechanical designer cannot build his side of the equation nor discuss his problems intelligently, unless he knows what factors must be considered.

IT IS gratifying to know that those electrical manufacturing executives who oppose informative technical publicity about their products are in the minority. Our best articles on motors and controls were contributed by the engineers of leading electrical manufacturers.

With all due deference to the objecting minority, we shall continue to edit and publish comprehensive and informative articles on motors and controls. That our readers want such information is evidenced by the many requests we have received for extra copies of articles that have appeared, from which it can be concluded that our readers derive profit from them. Publication of such articles is one method by which *Product Engineering* fulfills its objective of editorial publicity that contributes to the creation of more salable products. We stand pat on this objective.

Developments That Made The Motor Cars of 1935

The first part of this article (February number) dealt mainly with the materials used in the engine. This concluding installment sets forth developments in engine parts, transmission units, body trim and finish



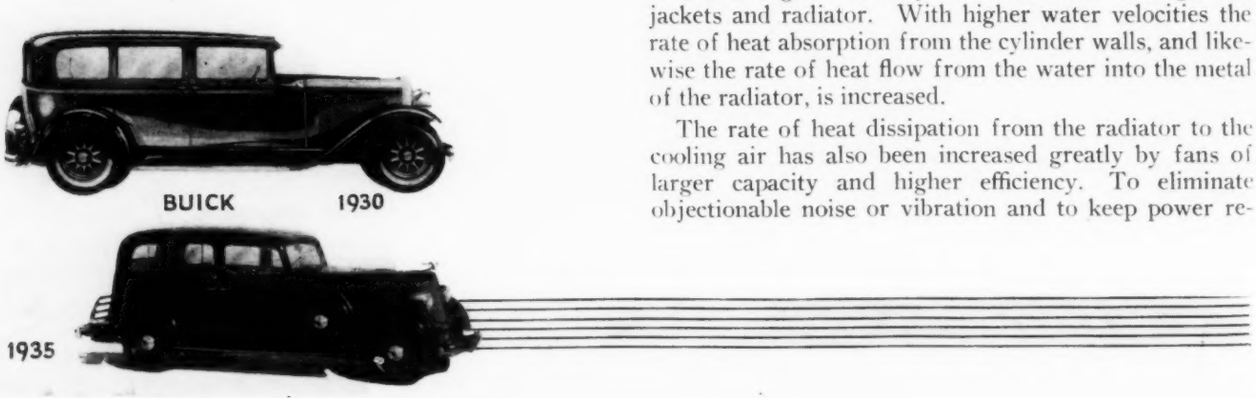
One result of the improvements in the design and manufacture of chain drives has been their increased use for timing drives

INCREASED engine horsepower and higher compression ratios have required changes not only in the engines themselves, but have also been the greatest factor compelling new developments in engine auxiliaries. Thus, cooling systems are now required to dissipate more heat, a design problem which has been further complicated by the style trend to an inherently less efficient radiator. Not only are the radiators higher and narrower but the flow of air is restricted by grills and inclosing shells.

Within the engine the problem has been to obtain a greater flow and a better distribution of cooling water through the cylinder heads and jackets, especially around the valve seats. A development of the past three years has been the use of copper-tube inserts in the castings to direct cool water against surfaces subjected to the greatest heat. Likewise, more attention is being given to cored channels to get better water circulation. Cylinder jackets in several 1935 passenger-car engines have been extended to the lower end of the cylinder bore to increase the flow of heat from the piston, a type of construction common to heavy-duty industrial engines. Similarly, the exhaust valve seats are entirely water jacketed in nearly all 1935 passenger cars, a standard practice for several years in truck and bus engines.

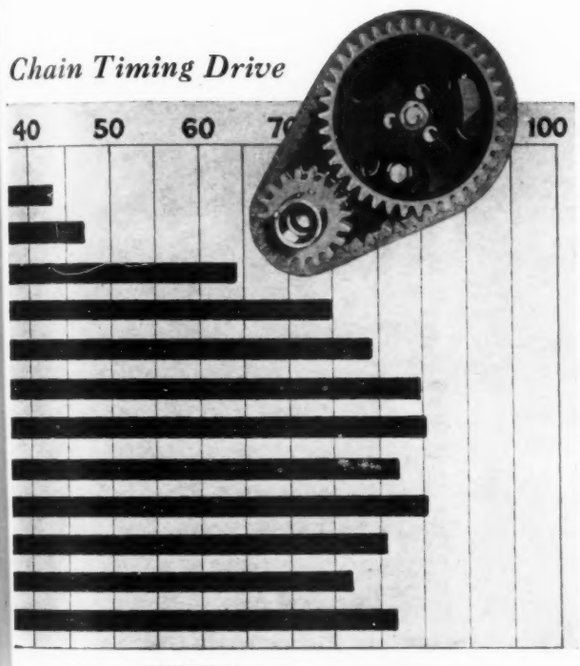
Greater heat dissipation with smaller radiators of less efficient shape has been obtained through larger pumps and larger fans. Over the past five years pump capacities in some cars have increased as much as fourfold. Only a small portion of this increase was necessary because of the added water jacketing. The major reason for the increased pump capacity has been to obtain a higher velocity of circulation through the jackets and radiator. With higher water velocities the rate of heat absorption from the cylinder walls, and likewise the rate of heat flow from the water into the metal of the radiator, is increased.

The rate of heat dissipation from the radiator to the cooling air has also been increased greatly by fans of larger capacity and higher efficiency. To eliminate objectionable noise or vibration and to keep power re-

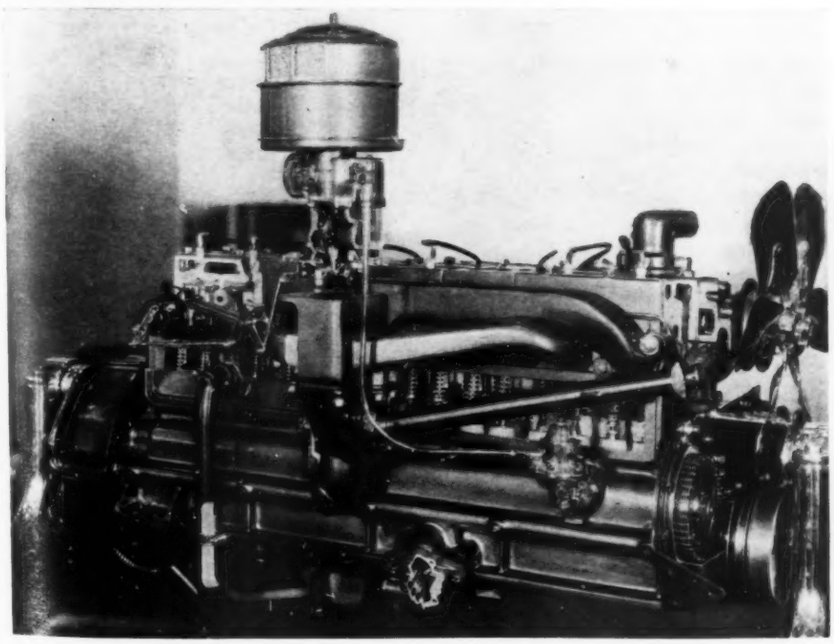


quirements low, features common to airplane propellers have been adopted by many designers of automobiles. Unequal angular spacing between the blades is commonly used for preventing resonant vibrations. On some of the larger cars, double fans are used.

With larger fans running at higher speeds, the necessity for better bearing performance has been met by the general adoption of needle, ball or roller bearings. V-belts are now used entirely for fan and pump drives for reduced slip and increased power at higher speeds.



Chain Timing Drive



And similarly, because of the greater bearing loads resulting from the increased pump capacity, the impeller shafts are also mounted on anti-friction bearings.

Chains have been favored for the camshaft drive, especially during the past nine years. According to some chief engineers of automobile companies, this trend is largely accounted for by the improvement in the performance of chains and sprockets. Of course, the cost factor is another important element, especially where relatively large center distances between camshaft and crankshaft require correspondingly large gears.

In response to the demand for cars that require a minimum amount of effort and knowledge to operate them satisfactorily, greater attention has been given to automatic controls, although not all the automatic equipment found on previous models has been retained. Automatic clutches, standard on some cars last year, are optional on 1935 models, while some automatic starting systems have been dropped or modified. This can be accounted for partly by the difficulty in perfecting automatic controls that will function properly under all possible conditions. However, for less complicated controls—for cooling water temperature, carburetor choke, spark advance and carburetor-mixture heater—automatic devices this year are standard on a larger number of cars than before.

Temperature control of the carburetor mixture was introduced in 1932. In its simplest form as applied to 1935 models, a bimetal thermostatic element in the intake tube to the carburetor serves as a vane. The vane when cold deflects intake air against the hot surface of the exhaust manifold. With higher intake air temperatures, the vane takes a position nearly parallel with the air stream deflecting less air against the hot spot.

In another design for temperature control of the carburetor mixture, a tube extends from directly behind the radiator fan to the carburetor. A bimetal coil located in the tube at the carburetor is thereby subjected to the temperature of the stream of air that has passed through the radiator. With engine and radiator cold, a butterfly valve under the

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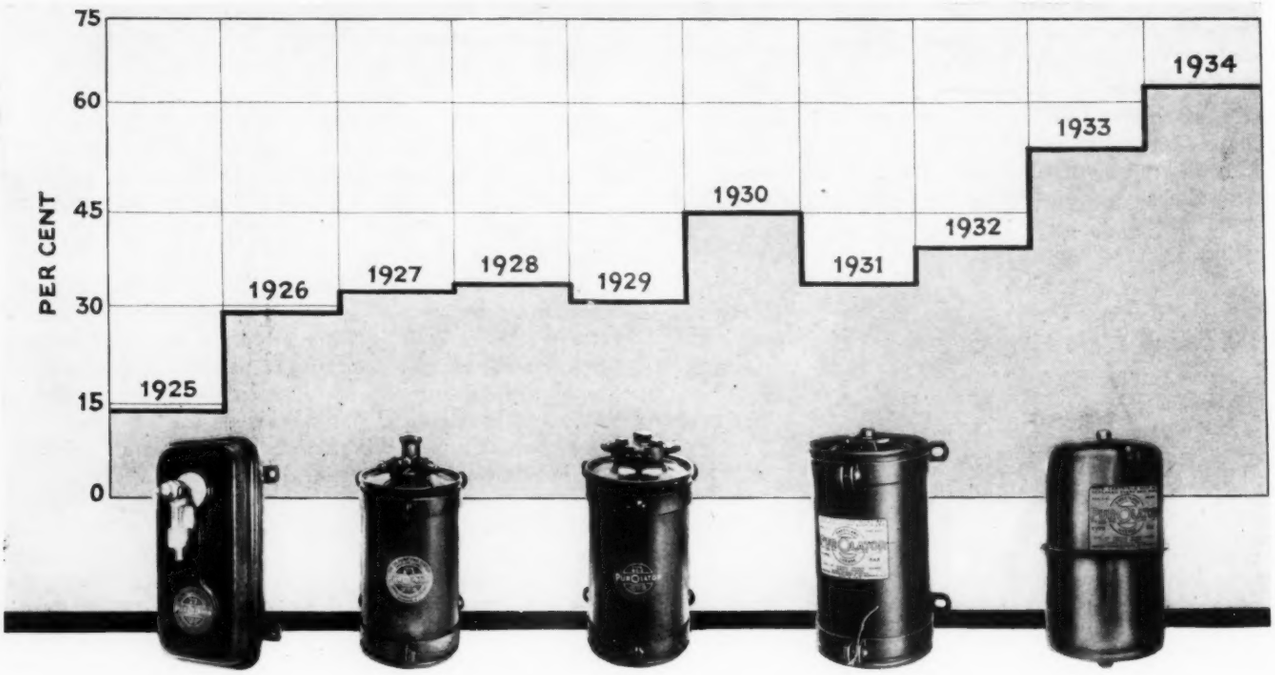
Automatic devices on the Pontiac motor regulate heat to the carburetor, choke valve position, water temperature and engagement of the starter pinion

Per Cent of Automobile Manufacturers Using Filters for Lubricating Oil



control of the bimetal coil directs some of the hot exhaust gases around the intake manifold, the intake mixture being heated by the exhaust gas. As the engine warms up, the increased radiator temperature heats the air passing through it, which causes the bimetal coil to deflect and thereby move the butterfly valve so as to cause less exhaust gas to be directed against the intake manifold.

Automatic chokes are also of a variety of forms. In one design a spring loaded bellows keeps the choke closed while the engine is cold and standing still. The bellows is connected to the intake vacuum through a bleeder, so that as soon as the engine starts, the bellows tends to contract and compress the spring inside, opening the choke. But the linkage controlling the position of the choke valve is also connected to a bimetal spiral which opposes the opening of the choke. When heated by the exhaust gas, the bimetal element deflects in the direction to open the choke. Thus the position of the choke is dependent on the intake vacuum and the exhaust gas temperature. When the engine has been stopped the choke closes partially but cannot close entirely until the exhaust manifold has cooled enough to allow the bimetal spiral to deflect back to the closed-choke position.

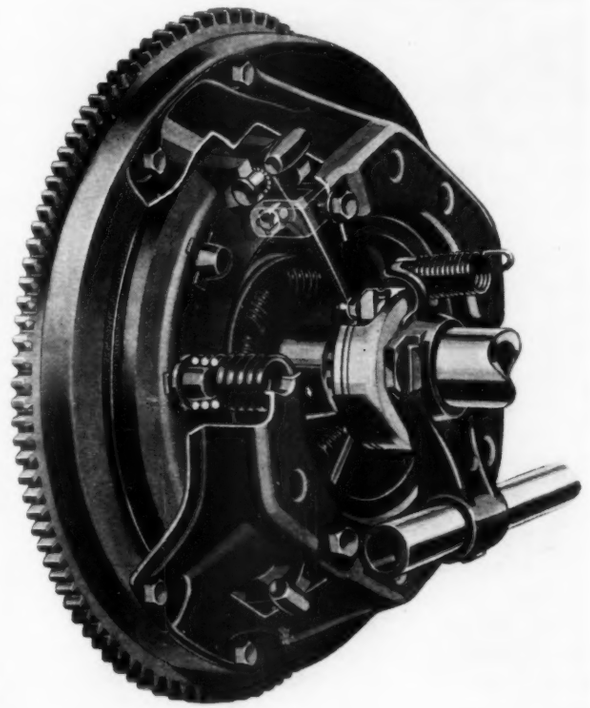
Another detail of this design is that the butterfly valve of the choke is pivoted off-center so that the flow of air tends to open the choke. Thus, there is a positive unbalanced force to take up the play in the linkages. To avoid the possibility of chatter or vibration, the linkage is connected to a dashpot.

In another common form of automatic choke, a bimetal spiral spring mounted on the exhaust manifold is connected through a linkage to the butterfly valve which is mounted with its pivot off-center. Flow of air through the offset choke valve tends to open it, but the

Water pumps with larger impellers carried on needle bearings increase rate of circulation in Hudson engines

valve is held shut by a bimetal spring when the motor is cold. A small piston actuated by the intake manifold vacuum also opposes the action of the bimetal spring when the motor is running, tending to open the choke.

When the motor is started the choke tends to be opened by the piston actuated by the intake vacuum and also by the unbalanced pressure of the air inrush against the eccentrically mounted butterfly valve. But until the engine warms up the position of the bimetal spiral prevents the choke from opening. After the engine has run a short time, the increasing temperature of the exhaust manifold deflects the bimetal spiral, the choke moving to its full-open position when the engine reaches normal operating temperature.



The Long clutch as used by Ford has centrifugal weights on the outer end of the clutch fingers to increase clutch pressures at high speeds

With this "triple control" arrangement the position of the choke is automatically established by the three factors—engine speed, accelerator position and engine temperature. The first two of these factors determine the vacuum in the intake manifold and the pressure tending to open the off-center butterfly valve. The third factor controls the bimetal spiral.

Several additional auxiliary controls, considered a few years ago as "gadgets," are now standard on many cars. Those cars having automatic choke also have "fast idling" to prevent stalling when the engine is warming up. This consists of a wedge-like element that holds the accelerator linkage in position for higher idling speed. The wedge is connected to the automatic choke linkage. As normal engine temperature is reached, the wedge element is withdrawn by the movement of the automatic choke.

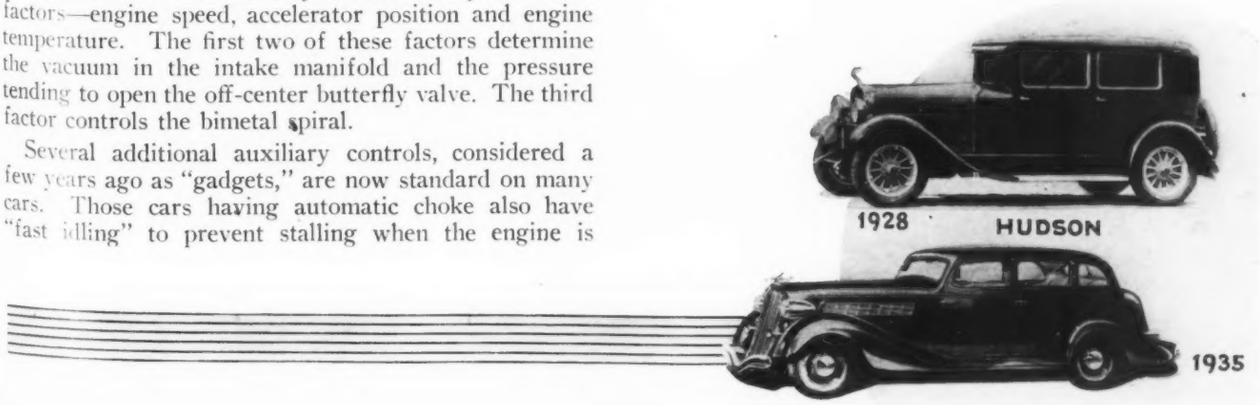
In some cars automatic controls are used to start the engine and for restarting the engine if it stalls. In one design the ignition switch key closes the circuit to a solenoid which closes the contacts for starting the motor. The starting motor circuit is opened only when the generator voltage rises enough to operate the generator cutout. A thermostatic time delay element in the starting solenoid circuit disengages the starter after one minute of operation.

On nearly all cars manual control of starting is greatly simplified by solenoid operation of the starting contacts and solenoids for engaging the starter pinion. The solenoid that throws the starter pinion into engagement is energized when the clutch or accelerator is depressed or is under the control of a switch on the dash.

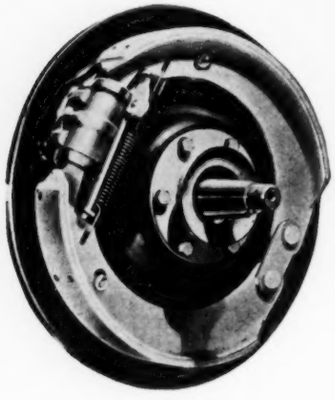
In all of these automatic starting systems, to prevent damage to the starter gear, the engine must be allowed to come to rest before the starter gear is thrown into engagement. Therefore, provisions must be made to prevent the energizing of the solenoid before the engine comes to rest when stalling or backfire occurs. In one design of safety device for this purpose, a vacuum switch on the intake manifold opens the solenoid circuit as soon as the engine starts, thereby disengaging the starter. When the engine stalls a small orifice between the manifold and the diaphragm of the vacuum switch allows air to bleed slowly into the vacuum chamber, thereby introducing sufficient time delay to allow the engine to stop before the solenoid can re-engage the starter.

Automatic controls are also used to maintain proper cooling water temperature by controlling the bypassing of the water around the radiator until the engine warms up. Recent trends have been to the use of bypass systems that do not restrict the flow of water through the cylinder jackets. In many of the 1935 models the bypass valve is under control of a bimetal or vapor-bellows thermostat and is located in a pipe external to the cylinder block, thereby avoiding complications in the cylinder jacket coring.

Gravity fuel systems have been entirely replaced by cam-driven pumps, and recently these pumps have been



Studebaker uses a larger diameter piston for the hydraulic brake on the front wheel. Rear brakes are both Servo and hydraulic



Looking at the under side of the new all-steel roof developed by Fisher Body. Stiffness is obtained from the dome shape and the welded-on ribs; drumming is eliminated by cemented fiber

given added functions. In several designs now in use, the fuel pump and a small vacuum pump are designed as one unit. The vacuum pump discharges into the intake manifold and furnishes a steady vacuum for operating the windshield wiper regardless of the manifold vacuum.

Many manual operations required in driving an automobile are now being studied for automatic control. Some progress is being made in automatic gear shifting, clutch operation and braking. One or more devices have been developed for each of these functions, but they are not yet in common use.

Automatic gear changing is being developed along three lines: (1) electrical control; (2) auxiliary clutches shifted by the centrifugal force of governor weights; (3) automatic hydraulic transmissions. In a few cars automatic clutch operation has been obtained by a combination of centrifugal action of governor weights and power derived from the intake manifold vacuum.

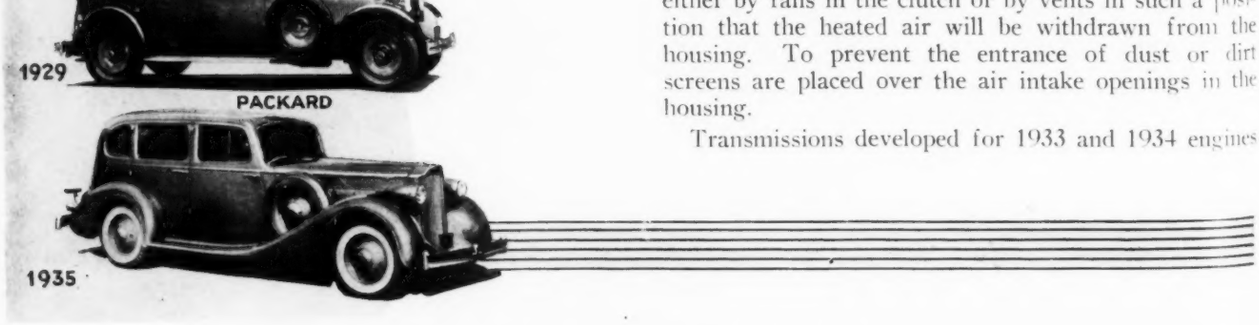
Nearly all clutches in 1935 models have reduced pedal effort, primarily through the obvious method of decreasing friction in the throwout mechanism. Clutch finger pivots and throw-out yokes are carried on needle bearings in several designs; knife-edge contacts rather than pin joints decrease friction in the foot-lever linkage. Some cars use sealed-in ball bearings for the brake pedal, clutch pedal and clutch-pedal shaft.

Increased speeds have also affected clutch design. To prevent excessive slip at high engine speed, greater unit pressures are required on the clutch facings.

Since pedal pressures are already high it is not desirable to increase clutch-spring pressure. In the design of the Long clutch, shown here, increased clutch pressure is obtained by the centrifugal action of small counter weights integral with the outer ends of the clutch fingers. At high speeds the centrifugal weights add to the pressure exerted by the springs. This design allows the use of even lower spring pressure, thereby requiring only a low pedal effort. At high speeds the centrifugal action builds up the clutch pressure to a value higher than could be obtained readily by springs alone. Needle bearings are used to carry the increased loads on the clutch finger bearings.

With the clutch housing proportioned to the motor dimensions, higher engine output with the same size clutch housing required some method of increasing the rate of heat dissipation per unit area of the housing. This was necessary because excessive temperature reduces the stiffness of the clutch springs, which tends to permit increased slip, thereby generating more heat. Nearly all clutches now have some form of air cooling, either by fans in the clutch or by vents in such a position that the heated air will be withdrawn from the housing. To prevent the entrance of dust or dirt screens are placed over the air intake openings in the housing.

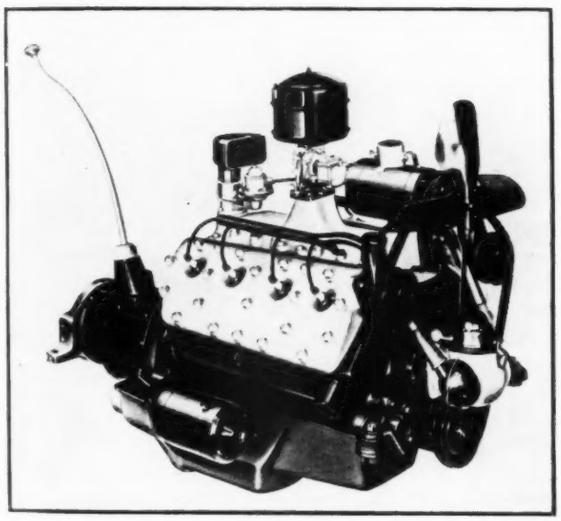
Transmissions developed for 1933 and 1934 engines



have been carried over into 1935 cars with comparatively little change. Nearly all cars now use helical gears with the countershaft gears in constant mesh. Reverse idler gears are also helical in some designs.

Gear materials in use today include plain nickel steel, nickel-chrome and nickel-molybdenum steels, with a trend toward greater use of the last alloy. Present methods of gear cutting and grinding give quiet, efficient drives that offer little need at present for extensive development.

The early attempt to provide complete chassis lubrication by the "one-shot" system and variations thereof has been abandoned almost entirely. It proved impractical to feed every bearing from a central lubricator and the necessity of leaving certain bearings to be given individual attention defeated the primary purpose of this lubricating system. It did not eliminate the necessity of lubricating service.



Fan and generator shaft on the Ford engine is carried on two Timken tapered roller bearings

The trend today is toward pre-lubricated bearings or oilless bearings, especially for places hard to get at or out of sight. Thus, pre-lubricated ball and needle bearings are being used for rear-axles, steering gears, clutch and brake pedals, clutch throw-out bearings and universal joints.

One lubrication development in which the automotive industry has led the way is oil filtering. The accompanying chart shows the rapid growth in the application of oil filters to automobiles. An interesting feature of this development, pioneered by the Motor Improvements, Inc., is the successive improvement in the design of their filter. In the early models the cans were made with clinched seams, similar to food cans. The latest models have drawn seamless cups. The filtering element for the automotive filters is a cloth mounted on a wire coil in a manner to present a large filtering area.

Cooling of the engine oil and the automatic control of its temperature is another development compelled by the incessant demand for higher speeds and power.

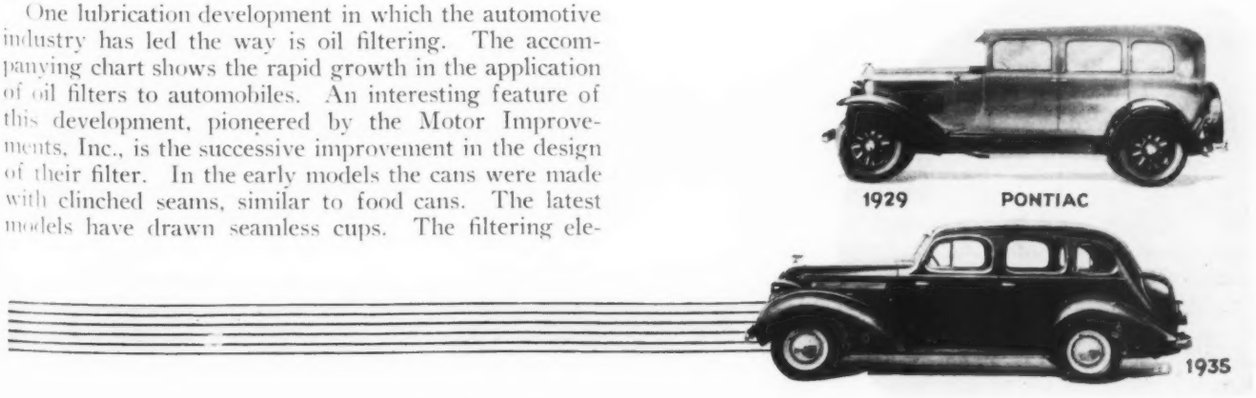
Tapered roller bearings are finding increasing applications in the new models. Steering knuckles, transmission main shaft and gears, differential pinion and ring gear, rear axle and front wheels, and the Ford fan and generator shaft are typical applications for these bearings in 1935 cars. Wherever large thrust loads are encountered, the tapered roller bearing seems to be favored.

Vibration absorbing engine mountings have now become standard on all cars. This development, brought about by improved methods of vulcanizing and better rubber, has been entirely successful. The degree of freedom allowed by the mounting is not the same for all designs.

Body mounting points are also insulated with live rubber to a greater extent than in previous models. Rubber spring shackles are also widely used to absorb high frequency vibration from tires and road shocks.

Not all parts of the car are subjected to the same degree of corrosion or possible destruction of the surface finish. Paint, enamel or lacquer alone is not sufficient to prevent rust. Several chemical rust-proofing processes have been adopted to an increasing extent during the past two or three years as the result of recent improvements in the process. One of the most important developments was a considerable reduction in processing time. In addition to forming a rust-proof surface, the process also offers an excellent paint base for any of the various types of finishes that are being used on the 1935 motor cars.

Indicative of the increased use of a rust-preventative undercoating for the paint finishes on automobiles, for the period from 1930 to 1934 inclusive, there was an increase of 66 per cent in the number of cars to which Parker Rust-Proofing Processes were applied, whereas during the same period automobile production dropped more than 50 per cent. Over the same period, based on units of rust-proofing energy consumed, the use of Parker Processes increased sixfold. In view of a drop of 50 per cent in the number of cars produced, this indicates that these processes are being used 12 times more extensively now than five years ago.



Stampings–III

Estimating Blanks for Drawn Parts

Methods for calculating the area and shape of blanks for pressed metal parts, with pertinent data and instructions, and typical examples to demonstrate the procedure

J. K. OLSEN
Formerly Chief Draftsman, Stewart Warner Company

THERE are various ways of calculating the size of the blank required to make a drawn part. But the most accurate and reliable method for obtaining the correct blank size is to cut-and-try—that is, to determine the size of the blank by actual operations in the press. Most die-makers and designers are familiar with this procedure. However, the cut-and-try method is unsatisfactory from a production standpoint. Usually it is desirable to have the size and shape of blank calculated from the drawings so that the raw material can be purchased without the necessity of waiting for the die to be finished. Probably the most practical and generally used rule for calculating the blank size is that the area of the blank must be equal to the area of the drawn part.

In calculating the size of the blank from the area of the drawn part, the theory is that the area of a blank does not change, regardless of the many different shapes the metal may assume when subjected to the drawing operations. However, in manufacturing it has been found that there is a slight difference between the area of the theoretical blank and the area of the finished part, caused by stretching of the metal. Sometimes the design of the die determines this slight difference and therefore it is generally not feasible to consider the stretch when figuring the size of the blank.

The logic of the method of figuring blanks from the area of the part can be seen from the example in Fig. 1, which illustrates general cases. Consequently this method can be used for drawn parts of any other shape, if properly analyzed. The accuracy of the method can readily be seen by proving it with a cylindrical shaped part. In a cylinder such as a piece of tubing, as shown in Fig. 2, wall thickness times mean circumference times length is equal to the amount of material in the part. The wall thickness and the length of the part are known. The mean circumference, called the neutral line, is the circumference based on the neutral diameter of the cylinder. This neutral diameter is $(O.D. + I.D.)/2$, the average between the outside diameter and the inside diameter of the cylinder.

To prove this is correct, consider the developed cylinder wall as in Fig. 2*A*. Right triangles *A* and *B* have one angle and the opposite sides, respectively, equal. Placing triangle *B* above triangle *A*, gives a rectangular section with a length equal to the circumference of the cylinder of a diameter equal to $(I.D. + T)$. The area of the material in the cylindrical wall is therefore equal

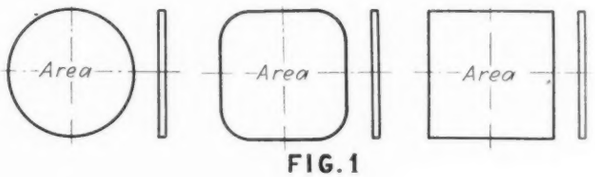


FIG. 1

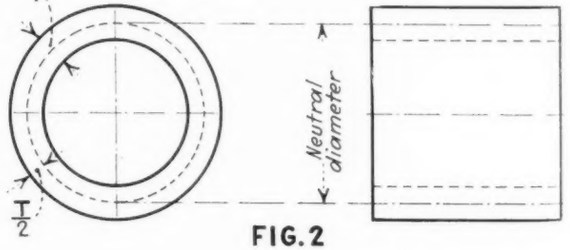


FIG. 2

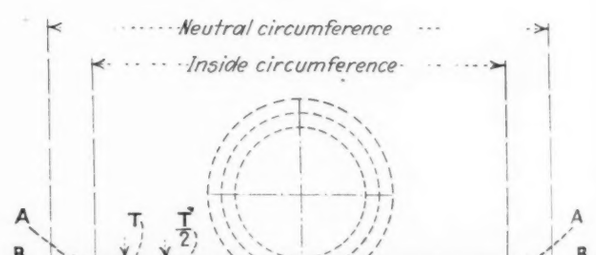


FIG. 2A

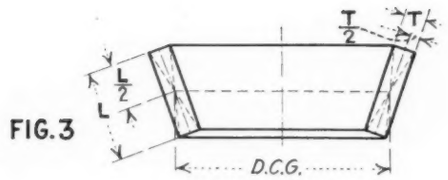


FIG. 3

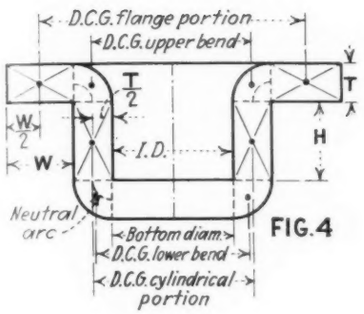


FIG. 4

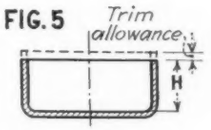


FIG. 5

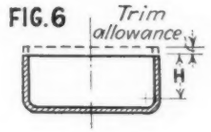


FIG. 6

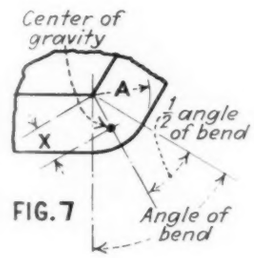


FIG. 7

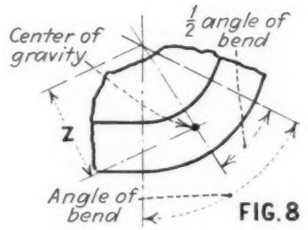


FIG. 8

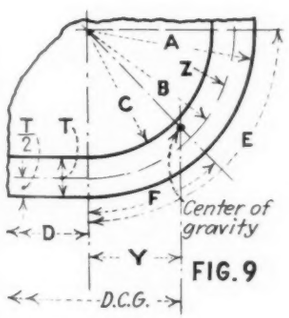


FIG. 9

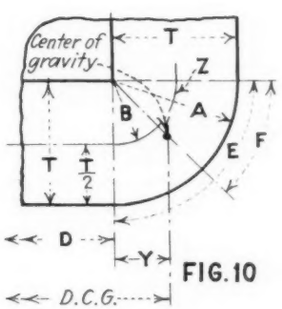
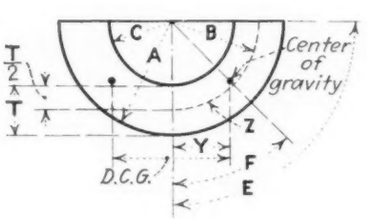


FIG. 10

FIG. 11



to: $\pi\ (I.D. + T)\ L$ where L = the length of cylinder.

Another proof that it is correct to use $\pi\ (I.D. + T)\ L$ when figuring the area of the material in a hollow cylinder, is as follows: Volume = (*area O.D.* — *area I.D.*) L. The volume divided by the thickness of the stock will give the area, and by dividing the area by the length, the neutral circumference or $\pi\ (I.D. + T)$ is obtained.

It would be correct to calculate the size of the blank from the volume of metal in the drawn part. Dividing the volume of metal by the thickness of the sheet would give the area of the blank. But this method is more cumbersome.

When the area of a cone-shaped part, Fig. 3, is calculated, a modified form of the second proposition of Pappus is employed, which, stated mathematically is: "The volume of any solid of revolution is equal to the product of the generating area by the circumference of the circle described by the centroid of the area," the centroid being the center of gravity. To avoid figuring the volume, the thickness factor is omitted from the proposition of Pappus, since the generating area of the work is the product of the length of the neutral line by the stock thickness. From this it follows that—the area of the required blank is equal to the product of the length of the neutral line and the circumference of the circle described by the center of gravity.

Let *D.C.G.* denote the diameter of the circle described by the center of gravity of the area of the section. The thickness of stock and the diameters are generally known from which the *D.C.G.* can be found. The area of the material required for this cone-shaped part can be calculated by finding the circumference of the circle described by the center of gravity and multiplying it by L, the length of the neutral line. The blanks for all surfaces which can be generated by revolving the cross-section about a center (surfaces of revolution) must be circular.

Trim Allowance for Cylindrical Sections

With the height H of the drawn part as in Fig. 5 or Fig. 6, the approximate trim allowance as given here should be added. All dimensions are in inches

H	Trim	*H*	Trim	*H*	Trim
1/4	3/64	1 1/2	7/64	3 1/2	11/64
1/2	1/16	2	1/8	4	3/16
3/4	5/64	2 1/2	9/64	5	7/32
1	3/32	3	5/32	6	1/4

When the cross-section is a simple geometric figure, such as a square or rectangle, the center of gravity is at the geometrical center. When, however, there is either a sharp or circular bend, the center of gravity of the curved portion can be found only by using special formulas, which will be discussed further on. Sharp corner bends or draws are objectionable, but often they cannot be avoided.

In Fig. 4 is shown a typical design of a cup for which the diameter of the blank can be determined easily by using the area method. There are five portions, each of which is calculated separately and then added together. They are as follows:

(1) The area of the flange portion = $D.C.G. \times \pi \times W$
(2) The area of the upper bend = $D.C.G. \times \pi \times$ *neutral arc*
(3) The area of the tubular portion = $D.C.G. \times \pi \times H$
(4) The area of the lower bend = $D.C.G. \times \pi \times$ *neutral arc*
(5) The area of the bottom = $\pi\ (I.D.^2/4)$

Adding these portions together the theoretical area of the part is obtained. However, another factor of im-

portance which must be kept in mind is an allowance of metal for trimming. There are only a few exceptions where no trim allowance is made, as, for example, where it is permissible for the drawn part to have a

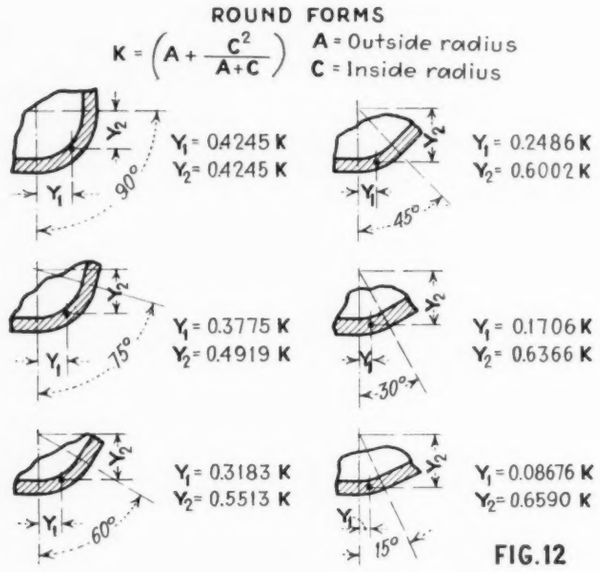


FIG. 12

rough edge and the affected dimension does not have to be held to close limits. The accompanying table can be used as a guide for trim allowances.

When figuring the blank sizes for drawn parts the trim allowance should be included, preferably in the cylindrical portion. Referring to Fig. 6 as an example:

(1) Area for tubular portion = *D.C.G.* $\times \pi \times$ (*H plus trim*)
(2) Area for bend = *D.C.G.* $\times \pi \times$ *neutral arc*
(3) Area for base = π (*Dia. of flat portion*/2)²
Area (1) + (2) + (3) = *Total area*
1.128 $\sqrt{\textit{Total area}}$ = *blank dia.*

There are various types of sharp corner draws of different angles and the raw material required for the

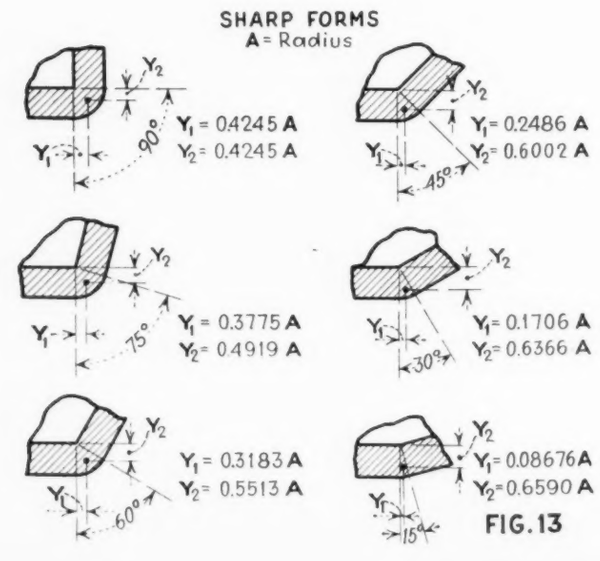


FIG. 13

bends can be determined by a method similar to the one used for 90 deg. bends. In this type of bend it is necessary to consider the angularity of the form and the location of the center of gravity. For any angle of bend the product (neutral radius $\times$ 0.01745 $\times$ deg. of bend) will equal the length of the arc on the neutral line.

The location of the center of gravity of a sharp bend from the center of the radius *A*, Fig. 7, equals

$$X = \frac{38.197\ A \sin (\text{angle of bend}/2)}{\text{Angle of bend}/2}$$

The location for the center of gravity of a circular bend from the center of the radii, Fig. 8, equals

$$Z = \frac{38.197\ G \sin (\text{angle of bend}/2)}{\text{Angle of bend}/2}$$

G = Outer radius + [(inner radius)²/sum of radii]

The following symbols are used for Figs. 9 and 10:

A = Outer radius	E = Angle of bend
B = Neutral radius	F = (Angle of bend)/2
C = Inner radius	T = Stock thickness
D = Inner diameter	$T/2$ = Stock thickness/2

Z = Length of neutral arc = 1.5708 B for 90 deg.
D.C.G. = $D + 2Y$
Area of material required = *D.C.G.* $\times$ 3.1416 $\times$ Z

In the rounded form, as illustrated in Fig. 9,

$$Y = \text{Offset} = \frac{38.197\left(A + \dfrac{C^2}{A - C}\right)\sin^2 F}{F}$$

In the square form, as illustrated in Fig. 10,

$$Y = \text{Offset} = \frac{38.197\ A \sin F}{F} \sin F$$

To calculate the hemispherical form, Fig. 11, the volume method is used, where:

A = Outer radius	= 2 in.
B = Neutral radius	= 1¾ in.
C = Inner radius	= 1½ in.
E = Angle of Bend	= 90 deg.
F = Angle of bend/2	= 45 deg.
$G = A + [(C^2/A + C)]$	
T = Stock thickness	= ½ in.
$T/2$ = Stock thickness/2	= ¼ in.

Z = Length of neutral arc = 0.01745 BE = 2.748 in.

$$Y = \text{Offset} = \frac{38.197\ G \sin F}{F} \sin F = 1.122 \text{ in.}$$

D.C.G. = $2Y$

Area of part = *D.C.G.* $\times \pi \times$ neutral arc = 19.373 sq.in. Checking by volume method, the volume of the large hemisphere = 0.5236 (diameter) $^3/2$ = 16.755 cu.in. and 7.069 cu.in. for the smaller hemisphere. The volume of the larger minus the volume of the smaller divided by the thickness of stock will give the area, which in this case equals 19.373 sq.in. In figuring blank sizes for drawn parts, the position of the curved section with relation to the centerline must be taken into consideration.

The tables, Figs. 12 and 13, for offsets Y_1 and Y_2 for frequently used angles will help to shorten the designer's calculations.

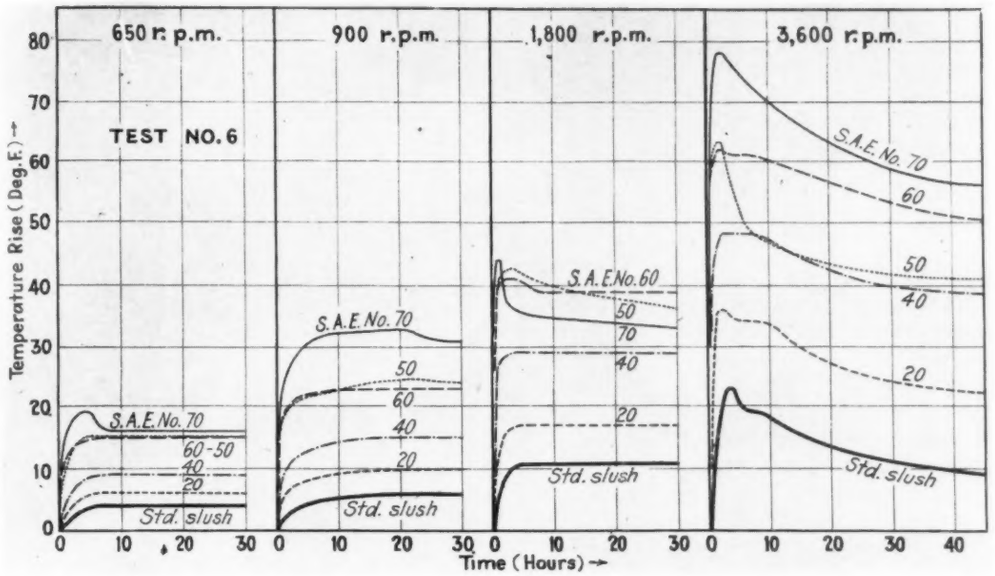


Fig. 5—Effect of oil viscosity and speed on the temperature of medium-size S.A.E. No. 310 ball bearing. Numbers on the curve are S.A.E. viscosities. Slush temperature is shown for comparison

Lubrication of Anti-Friction Bearings

F. LEISTER
President, The Bearing Appliance Company

Part III
Oil Lubrication

IN THE second of this series of articles (*February P.E.*) two generalizations were established with reference to the effects of the properties and quantity of grease on the running temperature of the ball bearing. Although the subject matter covered in the previous article is far from exhaustive as to grease studies, conclusions other than those specifically discussed may be drawn. For example, the comparison of slushed operating temperature has been previously noted. Comparing large size double-row and medium size single-row bearings, it is seen that the former, as might be expected, run warmer than the smaller bearings.

TEST NO. 6—Somewhat similar determination can be made with oil as a lubricant. While viscosity (consistency), time, bearing size and speed affect the temperature of oil-lubricated bearings as do the same factors in grease-lubricated designs, there is in addition the question of rate of feed. And this is subject to so many variations as to make an exhaustive study with oil almost impossible. Consequently, only brief reference can be made to some of the possible variations in oil lubrication. What may be expected when oil viscosity and speed are varied in medium size single-row No. 310 bearings is illustrated by the curves in Fig. 5. For ready reference, the S.A.E. viscosities are used, which are given in the accompanying Table I.

Separate runs were made on identical bearings using oils of different viscosities for runs up to 45 hours. The test set-up is pictured in Fig. 8. Again reference

Table I—Oil Viscosity Numbers

S.A.E. Viscosity Number	Viscosity Range, Saybolt Universal			
	Sec. at 130 deg. F.		Sec. at 210 deg. F.	
	Min.	Max.	Min.	Max.
10	90	Less than 120	—	———
20	120	Less than 185	—	———
30	185	Less than 255	—	———
40	255	———	—	Less than 75
50	—	———	75	Less than 105
60	—	———	105	Less than 125
70	—	———	125	Less than 150

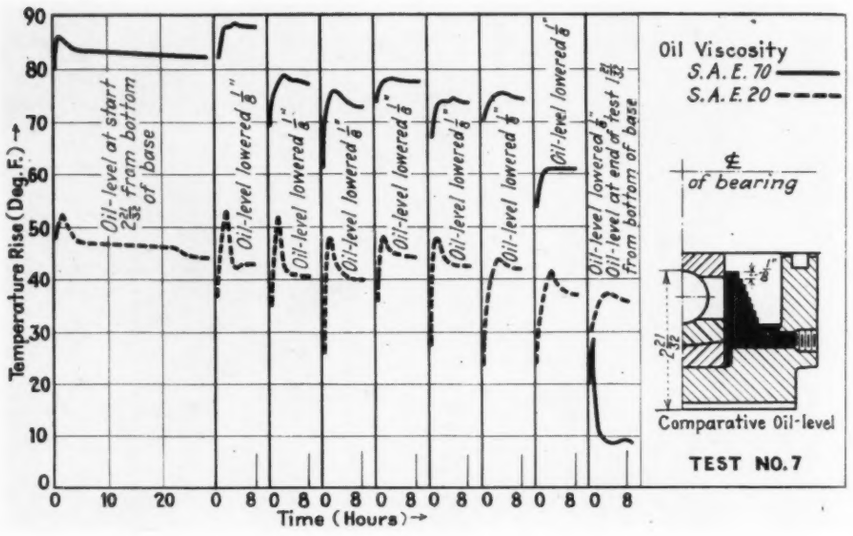


Fig. 6—Effect of oil viscosity and oil level on operating temperature, showing the trend to lower operating temperature with lower oil levels

is made to the slushed temperatures as a criterion. Each curve is labeled with the oil viscosity and the rise above room temperature is indicated by the ordinates. From this is concluded:

GENERALIZATION NO. 3—Bearing temperature may be expected to increase if speed or oil viscosity is increased.

This relationship is shown in slightly different form in Fig. 7. Temperature rise above room temperature for five S.A.E. numbers of viscosities is shown for speeds up to 3,600 r.p.m. and a curve is also shown for the standard slush condition for comparison. The bearings tested in Figs. 6 and 7 were bath-lubricated using side-level oil cups to maintain the level uniform during test, as can be seen in Fig. 8. Care was taken to keep quantities equal and, prior to the runs, the bearings were carefully operated and observed to be closely alike in all characteristics.

TEST NO. 7—For bath-type lubrication, the oil level is generally recommended to be in the neighborhood of the center of the bottom rolling element—ball or roller. The results of Test 7, plotted in Fig. 6, indicate what may be expected with reference to temperature rise when the oil level is above and below this point. The bearing used was a large single-row type No. 318. Tests at nine levels using two oils of different viscosities at each level were conducted in runs of 80 hours each at 3,600 r.p.m. The curves, therefore, give some idea of the difference to be expected in temperature with various oil levels and viscosities of No. 20 and No. 70 S.A.E., at this speed, 3,600 r.p.m. Each run was started with a carefully leveled-off supply and with bearing at room temperature. Starting at the high oil level, each run was completed successively and the oil level lowered $\frac{1}{8}$ in. after each run until the level was 1 in. below that of the first test. Between each run, all conditions were leveled off, so the initial start was under exactly the same conditions. This leads to the fourth generalization:

GENERALIZATION NO. 4—It is desirable to keep oil levels low.

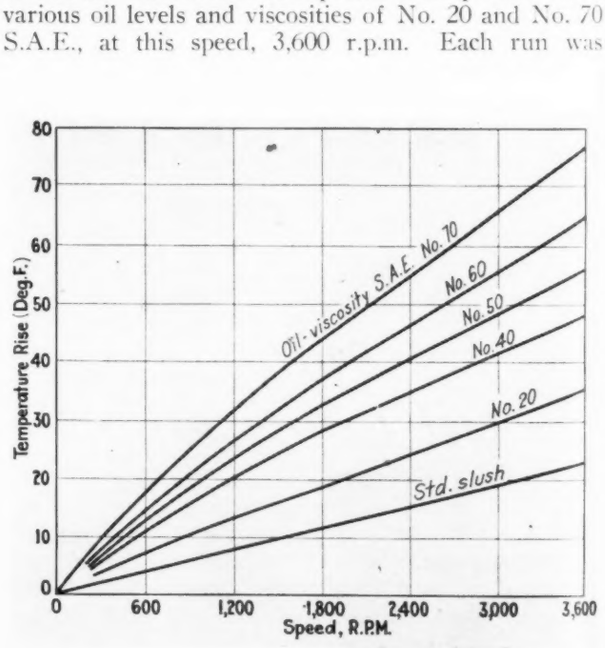
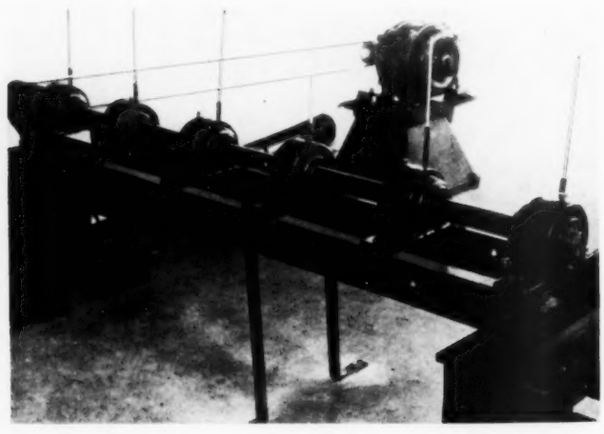


Fig. 7—Maximum temperature on a No. 310 medium series ball bearing at various speeds and oil viscosities

Fig. 8—Test apparatus for studying effect of varying viscosities and speeds on bearing temperatures. The thermometers extend down into oil sump



TEST NO. 8—On another large bearing set-up using No. 316 single-row bearings and maintaining oil levels at the center of the balls, readings indicated by the curves in Fig. 9 were obtained. Two viscosities, S.A.E. No. 20 and S.A.E. No. 70, were used, and the speed was 2,600 r.p.m. These bearings were actually under load. The curves show the initial temperature rise, and the subsequent gradual lowering of temperature resulting from the loss of oil by evaporation and thereby the lowering of the oil level.

TEST NO. 9—In Fig. 10 are shown the results of Test 9 to determine the temperature behavior with a drop-feed oiling system using extremes of oil viscosities, No. 20 S.A.E. and No. 70 S.A.E., and various drop feeds per minute at two speeds, 1,800 and 3,600 r.p.m. For comparison, the slush temperature is shown again. Each run was 50 hours. The bearing used was No. 310.

TEST NO. 10—In Fig. 11 are shown the results of Test 10 to determine the effect of load on temperature. This bears out the fifth generalization:

GENERALIZATION NO. 5—Within limits the magnitude of the load does not greatly influence the bearing temperature.

In all the tests reported here, the temperatures were measured from the oil sump in oil runs, and with the bulb against the outer ring in the tests on grease lubricated bearings.

Conclusion

It is seen therefore that different makes of grease, or different kinds of the same make of grease may cause varying bearing temperatures under otherwise identical conditions. For a few hours the temperatures may be high, but the bearing temperature gradually falls as the grease is expelled by the rotating parts. This initially high temperature should not be alarming unless it has been demonstrated that a number of hours of operation will not reduce the temperature. And the common mistake of adding more lubricant to reduce bearing temperature should not be made.

Increase in speed with the same grease, or increase in bearing size may cause higher operating temperatures. With the same speed and bearing size, greater quantities of grease will cause higher temperature.

With oil, identical bearings lubricated with higher viscosity oils or operated at higher speeds will result in higher temperatures. Further, higher levels and faster rates of oil feed will also increase bearing operating temperatures.

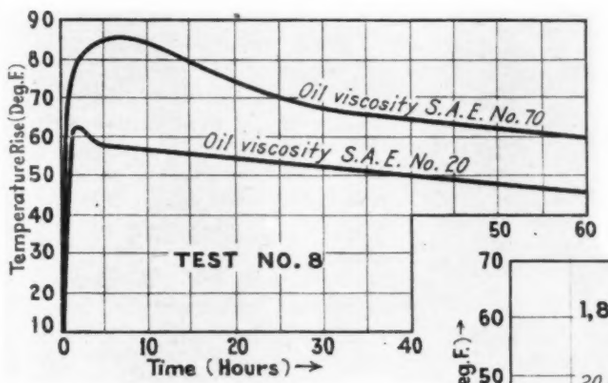


Fig. 9—(Above) Temperature rise in a fully-loaded bearing running at 3,600 r.p.m. with oils of different viscosity, and oil level curves and solid-line curves

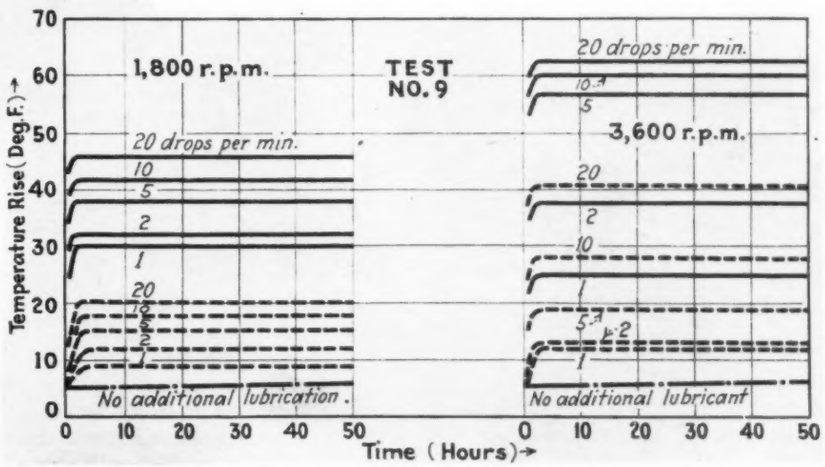


Fig. 10—(Right) Effect of various drop-feed rates with oils of No. 20 and No. 70 S.A.E. viscosity as shown by the broken-line curves and solid-line curves

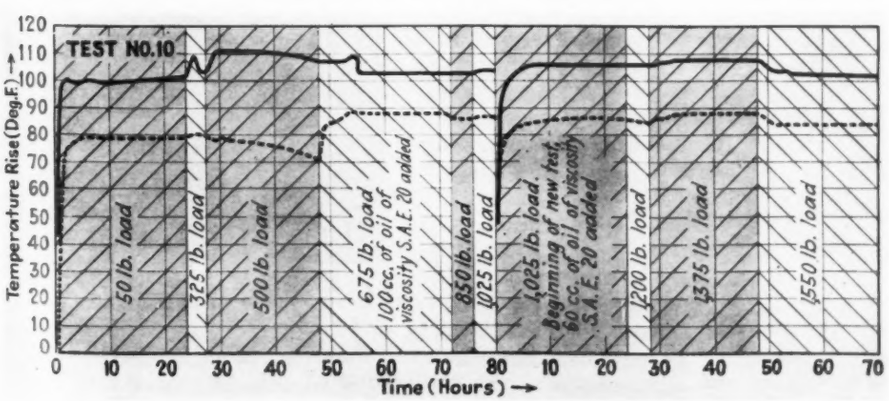


Fig. 11—Illustrating the small effect that load has on operating temperature at a speed of 3,450 r.p.m. Solid lines indicate temperatures using 70 S.A.E., and broken lines indicate 20 S.A.E. oil

PRODUCT DEVELOPMENTS

• *Teeterboard Motor Support*

Vertical slides on hand milling machines require some device such as springs and linkages or pulleys and counter-weights for counterbalancing the weight of the slide and motor drive. In the Whitney hand miller shown here, the weight of the motor itself is used to balance the weight of the slide. In this "teeterboard" construction the motor is mounted on the right-hand end of the pivoted casting as shown in Fig. 2. The eye in the left end of the casting is attached to the head slide. Because the center distance between the motor and the driven spindle is fixed, the tension of the V-belt is constant.

Fig. 1—In the Whitney hand miller, easy control of the vertical head is obtained by balancing the weight of the motor against the weight of the sliding head

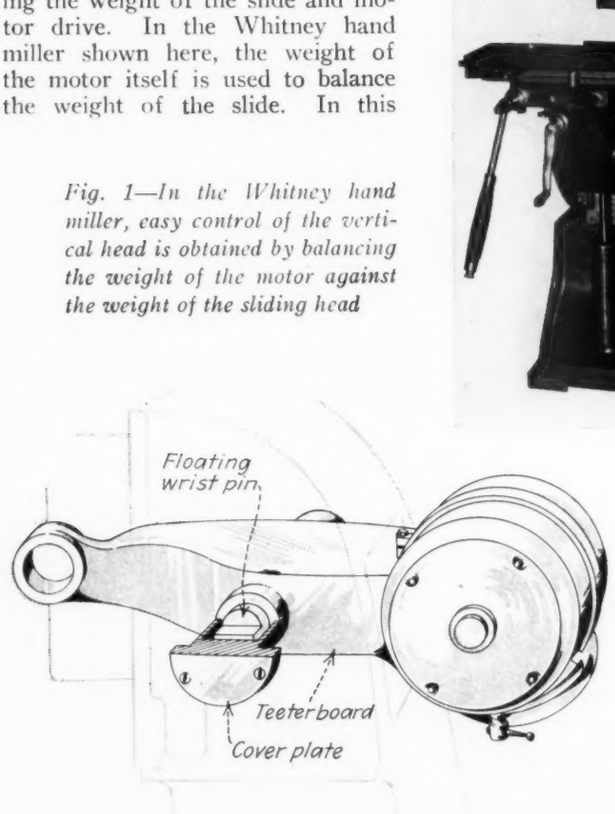


Fig. 2—Tangs in the wrist pin permit movement of the mounting arm relative to the machine frame to allow the eye to travel in a vertical straight line

Since the head slide travels in a vertical straight line, the point of support for the balancing arm can not be fixed in the machine frame. To allow a small horizontal motion at the "teeterboard" support the floating wrist pin has tangs on each end which slide in slots on the inner faces of two disks. These disks are bolted to the frame on each side as can be seen clearly in Fig. 1. The disks also bear against the sides of the pivoted casting, preventing lateral movement. The eye around the spindle bearing is faced and closely restrained, so that motion of the outboard motor is confined to a vertical plane.

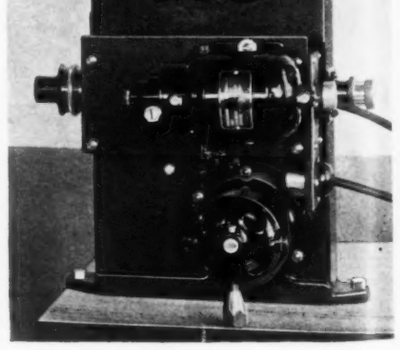
To adjust V-belt tension, the motor is rotated in its mounting, the eccentric position of the jackshaft changing the center distance between the drive pulleys and the spindle.

Both the table and dials on the knee are well illuminated by a built-in lamp supported on flexible tubing. With the gooseneck screwed into a pad in the column, the wiring from the lamp to the drum switch is entirely inclosed and protected. The small reflector can be placed near the work, supplying increased light intensity needed for precision machining.

• *High-Speed Camera*

Stroboscopes have been widely used in observing recurring phenomena. But it has not been possible to use stroboscopes to obtain photographic records with short time exposures because of inadequate light sources and camera equipment.

A new high-speed camera recently developed by the General Radio Company not only makes photographic records of recurring stroboscopic effects, but also photographs non-recurring or transient phenomena. By means of a low inertia film drive and motors of special characteristics, the camera attains a speed of more than 500 pictures per

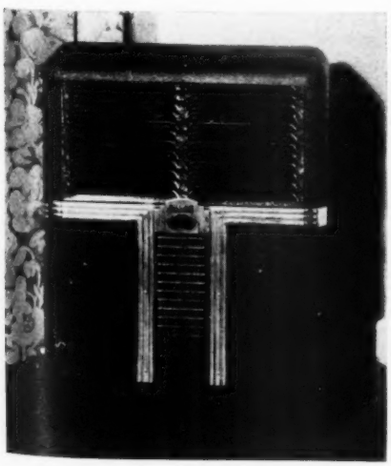


High-speed camera for use with a stroboscope to obtain photographic records of recurring or transient phenomena

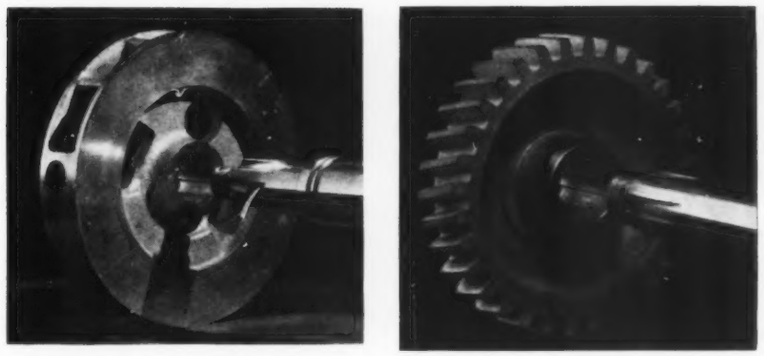
second, reaching full speed in less than one revolution of the film sprocket.

Adequate exposure at such high speeds is obtained by moving the film through the camera continuously, illuminating the focused area with a mercury arc lamp. By means of a commutator or electronic timer the lamp is flashed by a condenser discharge lasting but a few millionths of a second. Thus, for each flash a sharp exposure is made on the continuously moving film. To obtain a rapid acceleration, uniform film velocity and an adjustable speed, the camera is driven by two 115 volt universal motors mounted on the side of the housing. As shown in the accompanying illustration, the flange-mounted motor is direct-connected to the take-up reel. The other motor rotates the film sprocket within the camera through an 8.5 to 1 reduction gear built into the motor and a universal joint to avoid the necessity of lining up two sets of independent bearings.

The direct-connected motor tends to drive the take-up reel faster than the film is fed, eliminating any slack between the sprocket and take-up reel and takes care of the changing diameter on the take-up reel. To prevent overheating at low speeds the motors are run at reduced voltage. Film speeds are adjusted by controlling the voltages applied simultaneously to the two motors, giving film speeds from 1 to 35 ft. per second.



Built-in humidifier is a feature of the Nesco kerosene stove heaters, designed by Lurelle Guild. Finish is black satin enamel with chromium trim



Higher working stresses, more compact design and low machining cost are claimed for this new tapered spline developed by the Barber-Colman Company

• *Tapered Splines For Keying*

To prevent rotation between a hub and its shaft, two methods have been in common use: single keys for light load applications, and splined shafts or multiple keys for heavier loads. Because a single keyway causes a high stress concentration at the re-entrant corners of the shaft keyway, this construction requires the use of relatively low design stresses. On the other hand, although concentrated stresses are lower in splined fastenings than in equivalent keyed shafts, because of the distribution of the torque, high machining cost has prevented the more general use of splines.

A new type of tapered spline is now made commercially possible through the development of a new type of hobbing machine by the Barber-Colman Company. As shown by the illustrations, a tapered spline is hobbed into the shaft, and a similar spline broached into the hub, with the same amount and distribution of metal in each part.

The keys and the tapered root diameter are hobbed in the end of the straight shaft in much the same manner and with the same speed that gear teeth might be cut or straight splines be hobbed. A single cut is used to form the key sections with straight sides and parallel edges, as well as the tapered root diameter. Either four or six splines are used, depending on the size and strength required.

The greater shear area as compared with a single key gives several times the strength.

Contact area between the multiple keys of the taper spline not only gives greater strength but also greatly reduces stress concentrations. The keyways cut into the shaft are not so deep and being uniformly distributed around the section, result in a more uniform stress distribution. Since the keys themselves are integral with the shaft, they are not subject to twisting and distortion such as encountered by the conventional single key.

The cost of manufacture, according to Barber-Colman engineers, will be little, if any, more than for a plain single key, because of the simple hobbing operation on the new machine. But the greater strength of the taper spline construction will allow less metal in the shaft or hub in a given joint. The hub can be made lighter and shorter, and the shaft can be smaller in diameter.

For joints subjected to periodic reversals of stress, the tapered spline has the added advantage over a single key in having greater resistance to progressive distortion. Because of the low stress intensity, and uniform load distribution around the shaft, the tapered spline offers little opportunity for looseness to develop in the wheel mounting, such as might occur with a single key.

• *Die-Making Machine Features Built-In Light*

Designed as an integral part of the machine, a built-in fixture assures correct illumination for doing precision work on the new Foley die-making machine. In addition to careful attention to illumination, the design incorporates several other unusual features. Circular guides

supporting the table are centered at the upper face of the table allowing it to be tilted without enlarging the hole through which the saw or file passes. Likewise, the ram is adjustable, rotating around the center in line with the upper face of the table.

Die-cast, four-step pulleys on the motor and ram are driven by a V-belt, making possible four ram speeds. Ram stroke is also adjusttable by means of an eccentric.

Since the machine must necessarily be adjusted frequently to meet a wide variety of work, each adjusting screw has its own attached wrench. The hexagon head of each adjusting screw is turned down near the outer end, with a larger washer riveted to the nut. Except when needed, the wrench is retained in the turned groove, ready and in place when adjustments must be made.

• *Precision Bearings For Telescopes*

In the design of a new 82-in. reflection telescope for the McDonald Observatory, Timken tapered roller bearings are used in both the ascension worm gear and on the declination sleeve. These applications not only call for large diameters, but also fine precision. The first bearing, with a bore of 17 in., had a run-out of only 0.001 in., when tested without load. When mounted and adjusted under load, the run-out will be even less. The bearings for the declination sleeve are larger and of exceptional accuracy. The largest bearing has a bore of 43 in., and was delivered with a maximum run-out of only 0.0015 in. This bearing will carry the major part of both the thrust and radial load imposed by the 14-ton telescope.

• *Preheater For Food Products*

Developed for the preparation of food stuffs which must be preheated before being placed in cans, the Patterson continuous-feed Pre-Heater has a number of unusual design features. The outside of the jacketed shell is of mild steel; the interior may be mild steel, stainless, or other material as may be required for resistance against the corrosion effects of the food to be handled. Both cylinder and jacket are welded throughout.

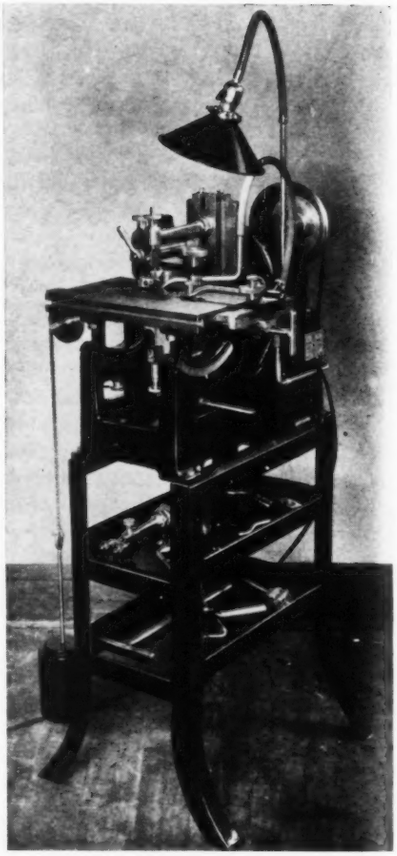
The machine operates with continuous feed and discharge, the load being gradually advanced toward the large discharge end by a series of paddles set at an angle and mounted on a central horizontal rotating shaft. The desired temperature in the machine is obtained by adjusting the pressure of the steam in the jacket.

One problem was to design the machine to thoroughly mix the ingredients without macerating them. It was also necessary to devise some means to prevent ingredients from adhering to the wall of the inner shell so that the food will not blanket the heat flow and will not scale from heating. For these reasons the interior of the shell is required to be smooth and, if necessary, as when made of hot-rolled carbon steel, it is machined.

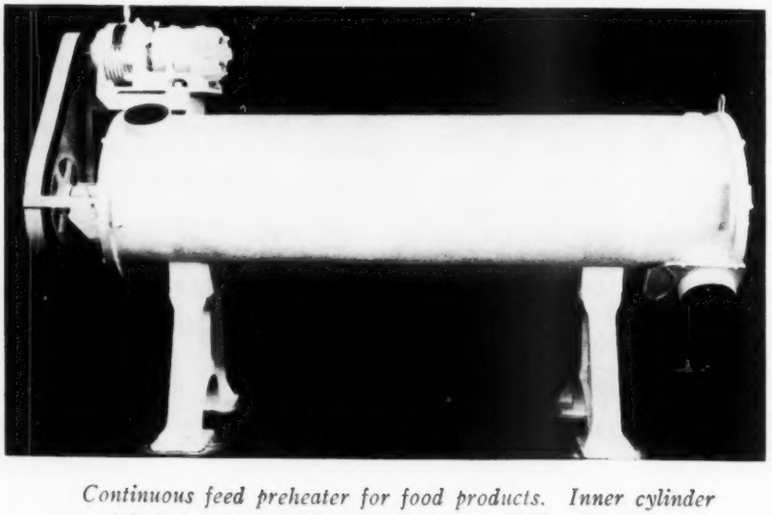
To keep the interior surface clean, one blade of each of the two-bladed paddles is equipped at its end with a spring-loaded scraper. The blade proper to which the scraper is attached is somewhat shorter than the plain arm opposite.

By loosening wing nuts, the head at the discharge end of the machine can be removed and the whole internal mechanism can be slid out, making everything accessible for cleaning.

Power is furnished by an electric motor coupled to a heliocentric type speed reducer with a chain drive to the slow-moving agitator shaft.



Built-in light on the Foley die-making machine has a flexible gooseneck and is adjustable for height



***Continuous feed preheater for food products. Inner cylinder** and jacket are arc-welded, bearings are of oilless type, drive is through a heliocentric type speed reducer*

The main bearings for the agitator shaft are copper-lead, oilless type, so that oiling or greasing, which might contaminate the food, will not be required. For the same reason, the bearings are placed outside of the stuffing boxes. The machine is finished in white Duco, a favored color for food machinery.

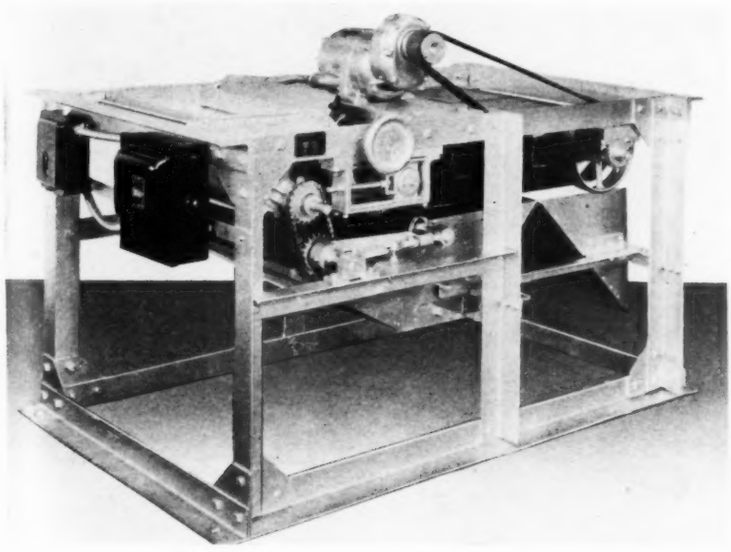
• *A.C. Magnetic Separators*

Finely ground materials are difficult to separate magnetically by the use of direct current because of the tendency of non-magnetic particles to cling to the magnetic material attracted to the magnet. For this reason the new Stearns magnetic separator shown in the accompanying illustration is designed to operate on alternating current rather than direct current, with a greater separating efficiency.

Material to be separated is fed to a horizontal conveyor belt. Above this belt and parallel to it is a similar one, the lower side of which moves in the same direction. Between the belts of the upper conveyor is an electromagnet energized by alternating current, thereby creating a strong magnetic field in the space between the upper and lower conveyor. As the material on the lower belt is being carried toward the discharge end, iron or magnetic particles are lifted against the face of the lower belt of the upper conveyor. Because of the nature of the magnetic field created by the alternating current, the particles are not held steadily against the underside of the belt but are vibrated with extreme rapidity, the agitation freeing the non-magnetic material from magnetic particles, regardless of the fineness.

At the discharge end of the machine, the magnetically held and suspended particles on the upper conveyor are carried out of the magnetic field, whereupon they drop off and into a chute. In the meantime the non-magnetic material carried on the lower conveyor is discharged into another hopper or chute.

The best speed at which to run the belt conveyors varies with the nature of the material handled. It is determined by laboratory tests and ranges from 75 to 150 ft. per min. The machine is driven by a spur-gear motor-reducer unit with the motor controls mounted on the machine, as shown in the illustration.



Pulsating current lifts magnetic material against the upper of two conveyors in the Stearns magnetic separator

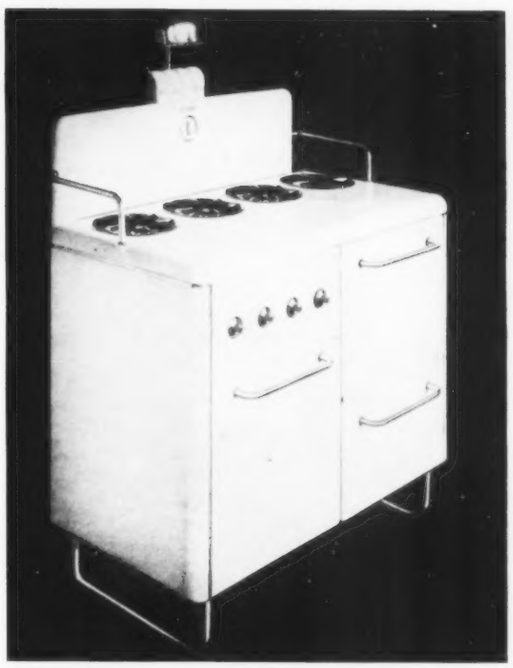
• *Arc Welded Range*

The entire inside body construction in the new Tinnerman gas range is built with all joints and corners arc-welded. Outside enameled sheets are fastened to the inside body through flexible clips, thereby preventing strain on the enameled sheets when the inside sheets expand or contract from temperature changes.

Greater utility is also claimed for the new range through the use of four burners in line across the back, this arrangement providing a large "table top" at the front for use in preparing food. Burner cocks are on the panel at the left, two of them equipped with simmer flame controls to make it possible to cook with the minimum use of gas. Oven heat is regulated by a Robertshaw bimetal thermostat. Below the valve panel is a utensil cabinet which has racks on the door.

All handles are tubular and chrome plated, designed extra wide so that they can also be used for hanging towels on to dry. The floor rests are of similar tubular construction, adjustable as to height, and like the door handles are finished in chrome plate.

A larger unit of this same model has the entire top and splasher panel made of Monel metal or stainless steel, whichever the purchaser may select.



Enameled sheets in this range carry no structural load, being held in place by flexible clips

Lateral Vibration of Shafts

This second of three articles, appearing in successive issues, gives equations for a shaft with three concentrated loads and explains the method of calculating the static deflections and coefficients

LEWIS F. MOODY
Professor of Hydraulic Engineering, Princeton University
Consulting Engineer, Baldwin-Southwark Corporation

IN THE first part of this article, which appeared in the February number, equations for the critical speed of shafts with one and two loads were derived. With three concentrated loads, that is, a shaft with three disks or wheels, as in Fig. 8, the same process may be applied, the reasoning for which will not be repeated. Assuming that one of the weights is slightly eccentric, the equations for the static deflections, to be used in determining the k's are:

$$y_s1 = k_{11}W_1 + k_{12}W_2 + k_{13}W_3$$
$$y_s2 = k_{21}W_1 + k_{22}W_2 + k_{23}W_3$$
$$y_s3 = k_{31}W_1 + k_{32}W_2 + k_{33}W_3$$

The meanings of the subscripts follow the same system as before, and as before the reciprocal theorem gives $k_{12} = k_{21}$, $k_{13} = k_{31}$, $k_{23} = k_{32}$, the order of the subscript numerals being indifferent. The equations corresponding to (10) and (11) are:

$$y_1 = (\omega^2/g)\,[k_{11}W_1\,(y_1 + e) + k_{12}W_2y_2 + k_{13}W_3y_3] \quad (18)$$
$$y_2 = (\omega^2/g)\,[k_{21}W_1\,(y_1 + e) + k_{22}W_2y_2 + k_{23}W_3y_3] \quad (19)$$
$$y_3 = (\omega^2/g)\,[k_{31}W_1\,(y_1 + e) + k_{32}W_2y_2 + k_{33}W_3y_3] \quad (20)$$

Eliminating y_2 and y_3, these equations give:

$$y_1 = \frac{e}{\dfrac{g/\omega^2}{k_{11}W_1 + (A + B + C)/[(D \times E) - k_{23}W_2W_3]} - 1} \quad (21)$$

$$A = k^2_{12}W_1W_3[(g/\omega^2) - k_{22}W_2]$$
$$B = k^2_{12}W_1W_2\,[(g/\omega^2) - k_{33}W_3]$$
$$C = 2k_{12}k_{13}k_{23}W_1W_2W_3$$
$$D = (g/\omega^2) - k_{22}W_2$$
$$E = (g/\omega^2) - k_{33}W_3$$

In Equation (21), y_1 becomes infinite when the denominator equals zero, and $\omega = \omega_c$, as before. If in the above values for A, B, D and E, the subscript c is used to indicate that ω has the value ω_c, and the denominator of Equation (21) is put equal to zero, simplification gives:

$$[(g/\omega^2_c) - k_{11}W_1]\,D_c \times E_c - k^2_{23}W_2W_3\,(g/\omega^2_c - k_{11}W_1) - A_c - B_c - C_c \quad (22)$$

This Equation (22) corresponds to Equation (13) for two loads. If, as before, we let $g/\omega_c^2 = y$, this equation may be expressed in a form corresponding to Equation (14), namely:

$$(y - k_{11}W_1)\,(y - k_{22}W_2)\,(y - k_{33}W_3) - k^2_{23}W_2W_3\,(y - k_{11}W_1) - k^2_{13}W_1W_3\,(y - k_{22}W_2) - k^2_{12}W_1W_2\,(y - k_{33}W_3) = 2\,k_{12}k_{13}k_{23}W_1W_2W_3 \quad (23)$$

In terms of the partial deflections this gives:

$$(y - y_{11})\,(y - y_{22})\,(y - y_{33}) - y^2_{23}(y - y_{11}) - y^2_{13}y_{31}\,(y - y_{22}) - y^2_{12}y_{21}\,(y - y_{33}) = 2y_{12}y_{23}y_{31} \quad (24)$$

This cubic equation can be reduced to the form:

$$y^3 - (y_{11} + y_{22} + y_{33})\,y^2 + (y_{11}y_{22} + y_{11}y_{33} + y_{22}y_{33} - y^2_{23} - y^2_{13} - y^2_{12})\,y = y_{11}y_{22}y_{33} + 2y_{12}y_{23}y_{31} - y_{11}y^2_{23} - y_{22}y^2_{13} - y_{33}y^2_{12} \quad (25)$$

In working out a practical problem, the numerical values of the partial deflection may be inserted at this point, and then the equation may be solved for y either analytically or graphically. (See Mark's Handbook, pp. 117-118, or Hudson & Lipka, Manual of Mathematics, p. 5.)

The direct solution of the cubic equation is rather laborious. Since we have available an approximate method of finding the critical speed, which will be explained later, giving in many cases a value very close to the accurate solution, the simplest method is to find an approximate value of y from the relation:

$$y = \frac{W_1y_1^2 + W_2y_2^2 + W_3y_3^2}{W_1y_1 + W_2y_2 + W_3y_3}$$

in which y_1, y_2 and y_3 are the total static deflections at

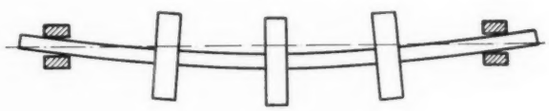


FIG.8

Fig. 8—Shaft with two bearings and three wheels or disks between them

each load due to all the loads, or:

$$y_1 = y_{11} + y_{12} + y_{13}$$
$$y_2 = y_{21} + y_{22} + y_{23}$$
$$y_3 = y_{31} + y_{32} + y_{33}$$

The above value of y may be inserted in Equation (24) as a first approximation and corrected by one or two successive approximations. Then from the final value of y:

$$\omega_c = \sqrt{\frac{g}{y}} \qquad N_c = \frac{187.6}{\sqrt{y}}$$

Determining the Static Deflections and the k Coefficients

In general, for cases where the static deflection curve is not known, it can be found either by successive integration or by "the method of area moments," as presented on p. 163 of Boyd's *Strength of Materials*. Space will not be taken here to review these methods. It may be helpful, however, to illustrate the determination of the k values in a few typical cases for concentrated loads. For more complex problems, and particularly for shafts of varying diameter or stiffened by hubs, as is the usual condition, either graphical methods or, still better, the use of models as developed by Prof. George E. Beggs will save time and complication. This last is described in the article, "The Use of Models in the Solution of Indeterminate Structures," published in the *Franklin Institute Journal*, 1927.

Case 1—Two Bearings With Two Concentrated Loads Between Them

Consider a shaft, as in Fig. 9, forming a simple beam between two bearings, and carrying two concentrated loads as shown, at points (*1*) and (*2*). We need to know the partial deflections, or those due to each load separately at each of the points. Applying first the load W_1 alone, the equation of the elastic curve to the left of (*1*) is known to be:

$$y = \frac{W_1 b_1 x}{6EIL}(L^2 - b_1^2 - x^2) \qquad (25)$$

To the right of load (*1*) the elastic curve is:

$$y = \frac{W_1 a_1 x_1}{6EIL}(L^2 - a_1^2 - x_1^2) \qquad (26)$$

In this equation $x_1 = L - x$. The deflection y_{11} at (*1*) due to W_1 alone, is given by Equation (25) with $x = a_1$, or:

$$y_{11} = \frac{W_1 a_1 b_1}{6EIL}(L^2 - b_1^2 - a_1^2)$$

$$k_{11} = \frac{y_{11}}{W_1} = \frac{a_1 b_1}{6EIL}(L^2 - b_1^2 - a_1^2)$$

The deflection y_{21} at (*2*) due to W_1, is given by Equation (26) with $x_1 = b_2$, or:

$$y_{21} = \frac{W_1 a_1 b_2}{6EIL}(L^2 - a_1^2 - b_2^2)$$

$$k_{21} = \frac{y_{21}}{W_1} = \frac{a_1 b_2}{6EIL}(L^2 - a_1^2 - b_2^2)$$

Applying W_2 alone, the equation of the elastic curve to the left of (*2*) is:

$$y = \frac{W_2 b_2 x}{6EIL}(L^2 - b_2^2 - x^2)$$

Putting $x = a_2$ we get:

$$y_{22} = \frac{W_2 a_2 b_2}{6EIL}(L^2 - b_2^2 - a_2^2)$$

$$k_{22} = \frac{y_{22}}{W_2} = \frac{a_2 b_2}{6EIL}(L^2 - b_2^2 - a_2^2)$$

Putting $x = a_1$ we have the deflection at (*1*):

$$y_{12} = \frac{W_2 b_2 a_1}{6EIL}(L^2 - b_2^2 - a_1^2)$$

$$k_{12} = \frac{y_{12}}{W_2} = \frac{a_1 b_2}{6EIL}(L^2 - a_1^2 - b_2^2)$$

This last equation is identical with the one for k_{21}, as already concluded from general considerations.

To obtain the critical speed it is only necessary to substitute in Equation (16) the k values found, or to insert in Equation (17) the partial deflection values.

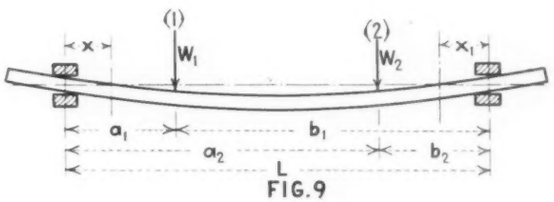


FIG.9

Figs. 9 and 10—The deflections y are measured from the axis of the bearings to the center line of the shaft

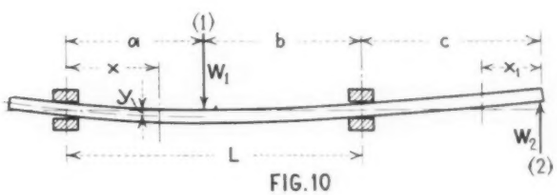


FIG.10

Case 2—Two Bearings and Two Loads One of Which Is Overhung

This case, shown diagrammatically in Fig. 10, corresponds, for example, to a hydraulic turbine-generator, two-bearing unit with generator rotor at W_1 and turbine runner at W_2, with either vertical or horizontal shaft.

Here W_1 and W_2 must be applied in opposite directions to produce the lowest critical speed.

With W_2 acting alone and calling downward deflections positive, the equation of the elastic curve between the supports is:

$$y = \frac{W_2 c x}{6EIL}(L^2 - x^2)$$

Then the deflection at (*1*) due to W_2 is obtained by

inserting $x = a$, giving:

$$y_{12} = \frac{W_2 ac}{6EIL}(L^2 - a^2)$$

$$= \frac{W_2 abc}{6EIL}(L + a)$$

$$k_{12} = \frac{abc}{6EIL}(L + a)$$

The equation of the elastic curve of the overhung portion, calling $x_1 = L + c - x$, is:

$$y = \frac{W_2}{6EI}[x_1^3 - (2L + 3c)cx_1 + 2c^2(L + c)]$$

By integration and by interchange of load and reaction and with y in the direction of W_2, that is, upward, taken as positive, and then putting $x_1 = 0$, the deflection at the load is given by:

$$y_{22} = \frac{W_2 c^2(L + c)}{3EI} \qquad k_{22} = \frac{c^2(L + c)}{3EI}$$

Deflections Due to W_1 Acting Alone

Deflection at the load W_1 is, as in the previous example of the single span:

$$y_{11} = \frac{W_1 a^2 b^2}{3EIL} \qquad k_{11} = \frac{a^2 b^2}{3EIL}$$

For the deflection y_{21} caused by W_1 at point (2) we have, as given above:

$$y_{21} = \frac{W_1 abc}{6EIL}(L + a)$$

$$k_{12} = k_{21} = \frac{abc}{6EIL}(L + a)$$

Case 3—Two Loads and Three Bearings With Loads in Different Spans

This is shown in Fig. 11 and represents, for example, a motor-generator set or a three-bearing pump and motor unit.

Deflections Due to W_1 Acting Alone

The reactions may be found from the equations:

$$R_1 = W_1\left(1 - \frac{c_1}{L_1}\right) - \frac{W_1 c_1}{2L}\left(1 - \frac{c_1^2}{L_1^2}\right)$$

$$R_2 = \frac{W_1 c_1}{2L_1^2 L_2}(L^2 - L_2^2 - c_1^2)$$

$$R_3 = -\frac{W_1 L_1 c_1}{2L L_2}\left(1 - \frac{c_1^2}{L_1^2}\right)$$

We can now proceed to determine the deflections. First remove the middle reaction; then the deflection at (1) due to W_1 alone acting on the simple beam of length L is:

$$\frac{W_1 c_1^2(L - c_1)^2}{3EIL}$$

Deflection at (1) due to R_2 alone, is from Equation (25):

$$\frac{R_2 L_2 c_1}{6EIL}\left[L^2 - L_2^2 - c_1^2\right]$$

$$= \frac{W_1 c_1}{2L_1^2 L_2}\left(L^2 - L_2^2 - c_1^2\right)\frac{L_2 c_1}{6EIL}\left(L^2 - L_2^2 - c_1^2\right)$$

$$= \frac{W_1 c_1^2}{12EILL_1^2}\left(L_2 - L_2^2 - c_1^2\right)^2$$

The resultant deflection with W_1 and R_2 both acting is the difference of these two values, since W_1 acts downward and R_2 upward, so that:

$$y_{11} = \frac{W_1 c_1^2}{12EIL}\left[4(L - c_1)^2 - \frac{(L^2 - L_2^2 - c_1^2)^2}{L_1^2}\right]$$

$$k_{11} = \frac{c_1^2}{12EIL}\left[4(L - c_1)^2 - \frac{(L^2 - L_2^2 - c_1^2)}{L_1^2}\right]$$

By the symmetry of the arrangement y_{22} and k_{22} are given by the same expressions by substituting W_2 for W_1, L_1 for L_2, L_2 for L_1 and c_2 for c_1, giving:

$$y_{22} = \frac{W_2 c_2^2}{12EIL}\left[4(L - c_2)^2 - \frac{(L^2 - L_1^2 - c_2^2)^2}{L_2^2}\right]$$

$$k_{22} = \frac{c_2^2}{12EIL}\left[4(L - c_2)^2 - \frac{(L^2 - L_1^2 - c_2^2)^2}{L_2^2}\right]$$

Following the same general procedure as above, the equations for y_{21} and k_{21} are:

$$y_{21} = \frac{W_1 c_1 c_2}{12EIL}\left[\frac{(L^2 - L_2^2 - c_1^2)(L^2 - L_1^2 - c_2^2)}{L_1 L_2} - 2(L^2 - c_1^2 - c_2^2)\right]$$

$$k_{21} = k_{12} = \frac{c_1 c_2}{12EIL}\left[\frac{(L^2 - L_2^2 - c_1^2)(L^2 - L_1^2 - c_2^2)}{L_1 L_2} - 2(L^2 - c_1^2 - c_2^2)\right]$$

The above examples illustrate the various methods of finding the partial deflections. The same methods will serve to work out more complex problems. Without describing the detailed development, the results will be given for two additional cases which will further illustrate the principles.

In the concluding article will be explained the case of a shaft with three bearings and two loads, one of which is overhung, and approximate methods of calculation.

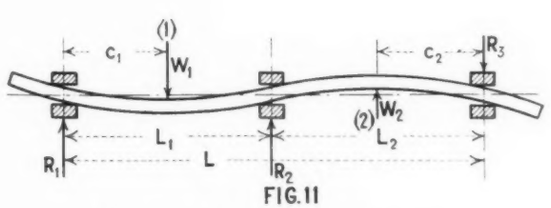


FIG. 11

Fig. 11—Representing a shaft with three bearings and a load in each span

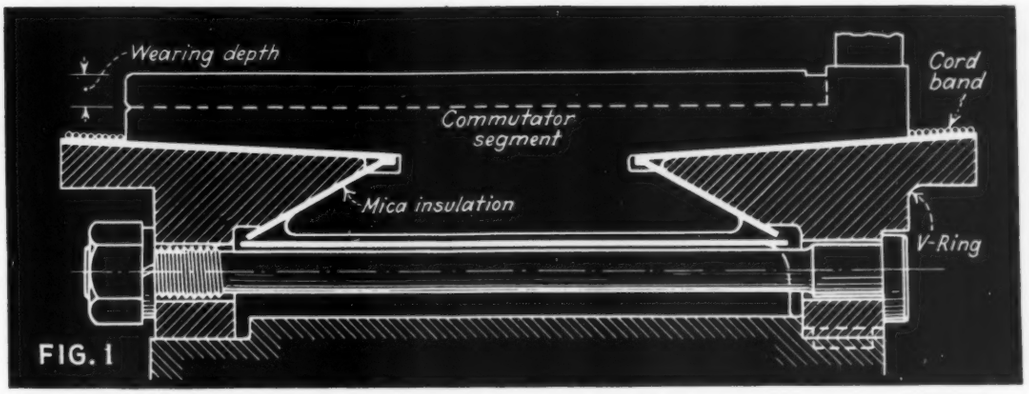


Fig. 1—Commutator segments are subjected to a combination of clamping stresses, centrifugal stresses, and thermal expansion effects

Stresses in Commutators

Although the procedure given here applies specifically to commutator design, the method has general application

J. R. ARNOTT AND C. S. DONALDSON

IN FEW mechanical designs are theory and practice so much at variance as in a commutator. In addition to withstanding centrifugal and clamping stresses, commutators are subjected to repeated heating and cooling, so that the design must allow the copper bars or segments to expand when heated and still be clamped rigidly when temperature returns to normal.

Because the stress distribution in a worn commutator differs greatly from that in a new one, maximum wearing depth must be determined. The initial stresses, for instance, will be distributed unequally in a worn commutator, while the tangential pressure between the bars, which keeps the segments from rising, cannot be the same. A satisfactory allowance for the radial wearing depth is given by the relation $0.041D + 0.1$ in. where D is the diameter of the commutator face in inches. A minimum allowance of $\frac{3}{8}$ in. is customary.

Considerations of the electrical design usually determine the diameter and the peripheral speed, but V-ring commutators cannot be economically built for peripheral speeds greater than 5,500 to 6,000 ft. per min. For ease of calculation, the following design work is based on a segment such as shown in Fig. 1, assumed to be 1 in. thick, measured circumferentially.

Referring to Fig. 2, a typical commutator segment, the lower limb is a short cantilever so that the line of fracture will be along some line *ab*, inclined to the line of action of the force producing the stress by an angle which can be assumed to be 15 deg. when $\theta = 30$ deg., the usual angle. Each limb carries half the centrifugal load with the addition of the clamping load. Assuming a clamping load equal to that exerted by the press when the commutator is being built, with this clamping load concentrated at the middle point of the actual supporting surface A, the bending moment M is given by the following relation in which W is the weight of a new segment 1 in. wide, N is r.p.m., R is the radius of gyration, and P is the clamping load in lb. per in. width of segment.

$$M = \frac{l}{2}(0.0000284\ WRN^2 + P \cot \theta)$$

With a section 1 in. wide of depth m, the bending modulus is $m^2/6$, and the maximum stress S in lb. per sq.in. is

$$S = \frac{6l}{m^2}\left(\frac{0.0000284\ WRN^2}{2} + P \cot \theta\right)$$

The quantity $0.0000284\ N^2$ is constant for any one commutator and its value, K, can be determined. If θ is 30 deg. the stress equation simplifies to:

$$S = \frac{3l}{m^2}(KWR + 3.46\ P)$$

If this stress does not give a factor of safety on the yield point of more than $2\frac{1}{2}$, the depth of the segment below the V-groove, and thus the value of m^2 must be increased. The strength of copper used for segments

varies to some extent with the size of the bars, but it can be assumed that the tensile strength of bars thicker than 0.20 in. does not fall below 40,000 lb. per sq.in., and that for bars less than 0.20 in. thick the tensile strength does not fall below 45,000 lb. per sq.in.

Stresses in the front overhang will be greater for worn segments than for new ones, and need be worked out for worn segments only. Where r is the radius and w is the weight of the overhang for a worn segment, the maximum stress of the section cd set up by the centrifugal force is:

$$S = \frac{3\ KRLW}{t^2}$$

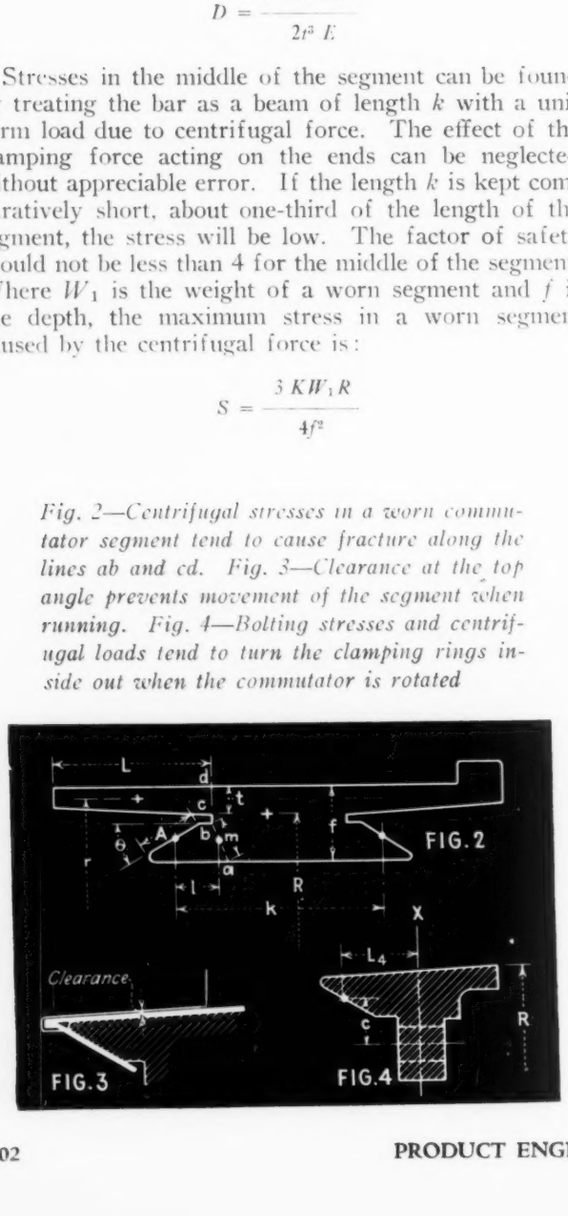
This calculation need be worked out for a worn segment only. The deflection at the tip of the overhang should not exceed 0.005 to 0.007 in., as any greater deflection causes an increase in the distance between the copper segments which may allow the mica spacers to rise. The depth t of the worn segment above the V-groove should not be less than 50 per cent of the depth m below the vee. Where E is the modulus of elasticity for copper, the deflection D is given by:

$$D = \frac{3\ KWRL}{2t^3\ E}$$

Stresses in the middle of the segment can be found by treating the bar as a beam of length k with a uniform load due to centrifugal force. The effect of the clamping force acting on the ends can be neglected without appreciable error. If the length k is kept comparatively short, about one-third of the length of the segment, the stress will be low. The factor of safety should not be less than 4 for the middle of the segment. Where W_1 is the weight of a worn segment and f is the depth, the maximum stress in a worn segment caused by the centrifugal force is:

$$S = \frac{3\ KW_1R}{4f^2}$$

Fig. 2—Centrifugal stresses in a worn commutator segment tend to cause fracture along the lines ab and cd. Fig. 3—Clearance at the top angle prevents movement of the segment when running. Fig. 4—Bolting stresses and centrifugal loads tend to turn the clamping rings inside out when the commutator is rotated



The load deflection D in inches is given by:

$$D = \frac{KW_1R\ k^3}{6.4\ f^3\ E}$$

In designing the clamping rings it is desirable to make the V-angle $\frac{1}{2}$ deg. less on the clamping ring than on the segment, as the copper pulls in approximately this amount when the clamping rings are tightened.

With an insufficient number of clamping bolts the part of the clamping ring between any two bolts will behave as a beam with an evenly distributed load. On the other hand, too many bolts, or excessive strength in the bolts may allow the segments to be clamped so tightly that they are compressed beyond their elastic limit, so that the clamping grooves open out. To localize permanent strains in the part where the least harm is done, smaller safety factors are used in the bolts than in the end rings, and the bolts are turned down on the unthreaded portion.

In addition to the clamping load, the bolts carry loads set up by the centrifugal force on the V-rings. The total centrifugal force P on each vee is given by the following relation in which t_1 is the thickness of the mica segment, and t_2 is the thickness of the copper segment.

$$P = \pi KWR^2\left(1 - \frac{t_1}{t_2}\right)$$

The total stress on the bolts, where A is the cross sectional area of the bolt, is given by:

$$S = \frac{\pi R}{A}(P + KWRT\tan\theta)$$

While a safety factor below 4 should be avoided, the stress in the bolts should not be less than the stress in the end rings.

Centrifugal load from its own mass and bending moments caused by the centrifugal force of the segments and the pull of the bolts, impose stresses in the end rings. The combined load tends to turn the ring inside out. Centrifugal stress in the steel end ring, from its own mass, is $0.00116\ r^2N^2$ lb., where r in this relation is the radius of the end ring periphery. Referring to Fig. 4 the bending moment M arising from the centrifugal force on each vee is:

$$M = \pi KWR^2L_4$$

To this must be added the total bending moment caused by the bolts:

$$M = 2\pi RPc$$

The maximum bending stress in the ring is the total bending moment divided by $2\pi Z$ where Z is the section modulus about the axis N, Fig. 4. The total stress is therefore,

$$S = \frac{R}{2Z}(KWRL_4 + 2Pc) + 40.8\ KR^2$$

It is seldom that further calculations need be made. Since stresses set up by thermal expansion can be figured only on assumptions that may not present a true picture of conditions, this factor cannot be evaluated by analysis.

Water-Lubricated Rubber Bearings

F. L. HAUSHALTER
Development Engineer
The B. F. Goodrich Company

LUCIAN Q. MOFFITT
President, Lucian Q. Moffitt, Inc.

The use of water as a lubricant necessitates a bearing material possessing a low friction coefficient. Molded rubber, incased in a metal housing, supplies such a need

FIG. 1

FIG. 2

Typical examples of rubber bearings for marine service. Fig. 1 illustrates a flanged shell type. Fig. 2 shows stern casting with scoop at left to admit water. Cross-section illustrates an incased rubber bushing with water lubricating grooves

WHEN it is difficult, if not impossible, to seal the bearing against entrance of water or other liquids, or in such installations where it is convenient to lubricate with water, this type of bearing, having a core of tough, resilient rubber compound, has many applications. As the coefficient of friction of wet rubber on metal ranges, at high speed, from 0.010 to 0.005, wear is consequently reduced to a minimum. This has been found especially true on stern and strut bearings for ships' propellers. These bearings are also used on centrifugal pumps, at various points in dredges, in hydraulic turbines, for deep well pumps and in the paper, mining, and sand and gravel industries.

The rubber is vulcanized to a metal shell in a mold under high pressure in order to get a firm bond between the two materials. Adhesions as high as 600 lb. per sq.in., can be obtained between the rubber and shells of bronze, steel, cast iron, Monel metal, and various other alloys both ferrous and non-ferrous. Usually the rubber will tear before the bond lets go.

These bearings are molded to size in the same manner as an automobile tire, the dimensions of the molds being carefully calculated to allow for the shrinkage in the rubber, so that when the bearing comes from the mold it is automatically provided with the proper clearance for the particular size of shaft for which it is intended. The shrinkage of the rubber compounds used in bearings varies from $\frac{5}{32}$ in. to $\frac{7}{32}$ in. per ft., and close control is kept on the processing of these compounds of rubber in order to make sure that the shrinkage is constant.

Where sand and grit particles are encountered, the rubber compound used in the bearing is similar to that on a pneumatic tire tread, for it is especially compounded with ingredients that make the rubber highly resistant to cutting and abrasion. It is a firm rubber which deflects but slightly under the unit pressures used on bearings of this type. Where problems of shaft vibration are encountered bearings of a softer rubber than the above are often used with beneficial results. What is normally termed hard rubber would easily support the load, but it would be too hard for particles of sand to imbed themselves in it and it would cut away just about as readily as babbitt or bronze. The rubber must be firm enough to carry the load without excessive deflection and yet soft enough to allow abrasive particles to be embedded in it.

Two types of rubber bearings are manufactured,

bearings with straight longitudinal flutes, and bearings with spiral grooves. The fluted type bearing is used on practically all applications both horizontal and vertical. Bearings with spiral grooves are, as a rule, used only on special applications, and where the shafts are vertical.

Longitudinal flutes are provided primarily as a means of lubrication, and to allow sand and grit to pass readily through the bearing. The bearing area has the edges well rounded to eliminate any wiping action that would tend to keep the water away from the rubber at the center of the bearing face. A great deal of experimental work has been done in order to determine the correct taper on the edges of the bearing surfaces, for under load these surfaces are deformed slightly and the angle of approach between the bearing surface and the shaft changes, producing a somewhat similar effect to the pivoted bearing pads found in a modern thrust bearing of the Kingsbury type.

The size of the flutes has been determined more or less empirically, as a compromise between obtaining the maximum bearing area and insuring sufficient lubrication for the bearing, as well as obtaining the proper shape of the flute to take care of sand and grit. It has been found that even in bearings up to 15 in. and 20 in. in diam., the width of the flute should never be greater than $3\frac{1}{2}$ to $3\frac{3}{4}$ in. Experience has demonstrated that more and shorter faces are to be desired rather than few faces with wide areas.

The thickness of the rubber wall varies with the diameter, and the type of service in which the bearings are used. In propeller shaft bearings, insurance underwriters demand that bearings on shafts not exceeding 9 in. in diam. shall be replaced after they have worn down $\frac{1}{4}$ in.; $\frac{5}{16}$ in. for shafts 9 in. to 12 in. diam., and $\frac{3}{8}$ in. on shafts over 12 in. diam. On other bearings, such as those used on vertical shafts of hydraulic turbines, where rubber bearings up to $31\frac{1}{2}$ in. in diam. are successfully used, the bearing is usually removed when worn as little as $\frac{1}{16}$ in. The wall thickness of rubber bearings is usually determined by the wear permitted by a certain predetermined condition, plus the depth of the water groove required to insure a positive flow of water through the bearing.

The allowable loads on rubber bearings are governed to a large degree by the shaft speeds, and the method of providing the necessary lubrication. As a rule, at speeds as low as 18 to 20 r.p.m., rubber bearings are not recommended unless the pressure on the projected bearing area is held under 25 lb. per sq. in. Rubber bearings operate successfully when shaft speeds are as low as 100 ft. per min. and where the load is 50 lb. per sq. in. of projected area. Many rubber bearing applications are in the marine field on propeller shafts, both large and small, and it is seldom that the load on these bearings, even on the largest ocean-going vessels, is greater than 30 to 32 lb. per sq. in. of projected area.

Range of Coefficient of Friction

At slow speeds and light loads, a metal bearing has a slightly lower coefficient of friction than a rubber bearing, but at high speeds, the friction of a rubber bearing is less than that of a metal bearing. The coefficient of friction of a rubber bearing at high speeds ranges from 0.010 to 0.005, as determined by tests.

In marine service, the ratios of length to diameter for rubber bearings are practically the same as for lignum-vitae, bronze or babbitt, and it is nearly always

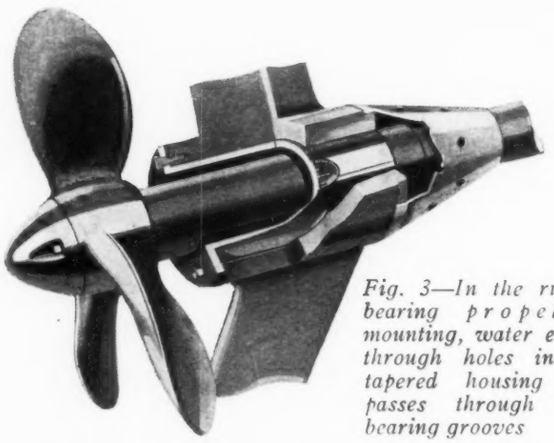


Fig. 3—In the rubber bearing propeller mounting, water enters through holes in the tapered housing and passes through the bearing grooves

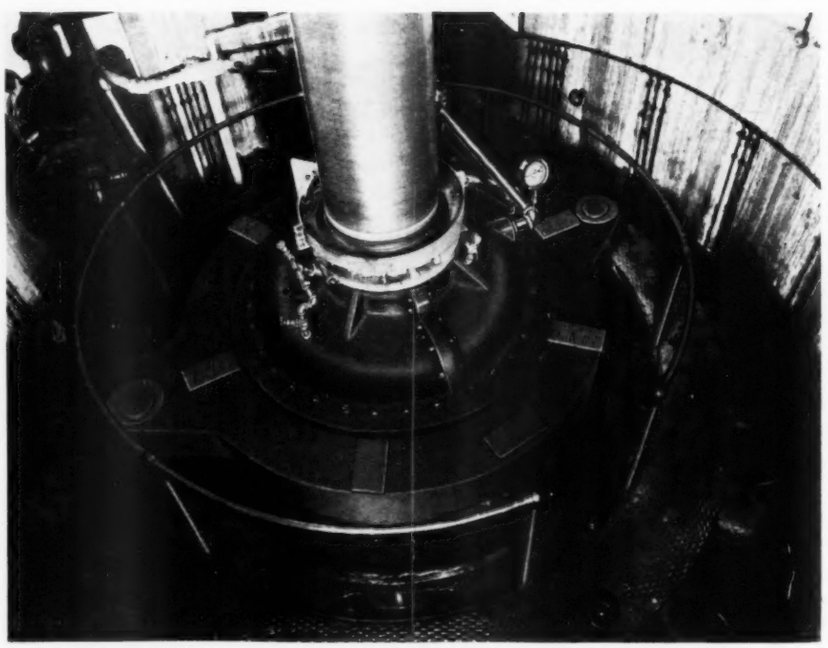


Fig. 4—In this 16,500 hp. hydraulic turbine the shaft has a bronze sleeve running in a 24 in. diameter rubber bearing which has been in service for more than ten years

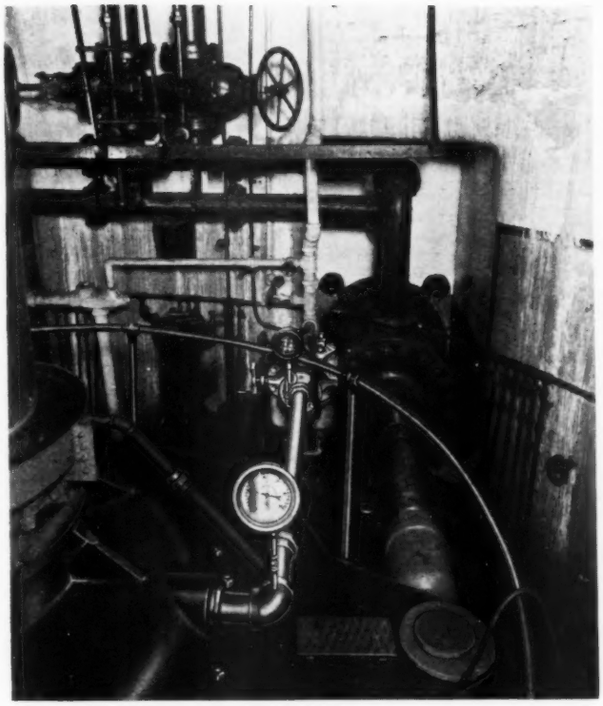


Fig. 5—Close-up of turbine in Fig. 4, showing the lubricating water supply to the bearing and the duplex strainer

possible to replace hard surfaced bearings with rubber without making any change whatever in size. The same applies to bearings for sand and gravel pumps, deep-well turbine pumps, hydraulic turbines, cutterhead bearings on dredges, and other services where hard surfaced bearings are used with water lubrication.

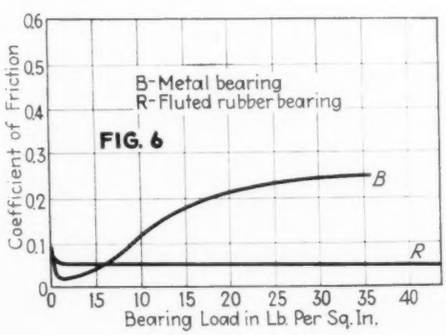
The deflection of a shaft in a rubber bearing under recommended loads is not as great as might be expected. For the bearing 19.685 in. diam. and 99 in. in length, the shaft deflection from no load to full load is 0.028 in. under the normal operating pressure of 25 to 30 lb. per sq. in. of projected area. This deflection is negligible in view of the fact that with rubber the clearance of 0.028 in. is maintained between shaft and bearing area over a long period of time because of the small amount of wear on the rubber.

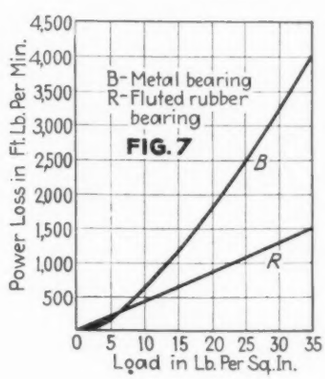
For hydraulic turbine shafts up to 31½ in. diam. it is not unusual for shaft oscillations to be held within 0.006 in. in amplitude. These shafts will run for years before the oscillation amplitude increases to as much as 0.012 in. even when the lubricating water contains abrasive material.

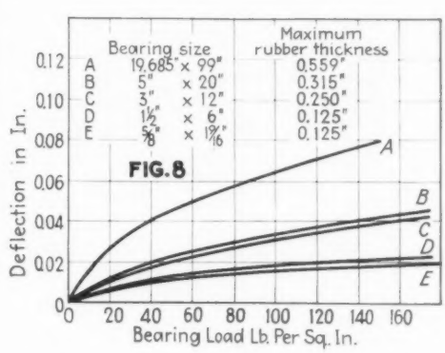
Numerous other advantages are claimed for rubber bearings. Many of these bearings have been in operation during the past several years on marine propeller shafts, hydraulic turbines, and other classes of service, where they have outworn hard surfaced bearings as much as 12 to 1. A fair average is considered to be 3 to 4 times longer life. Where abrasive particles are encountered in water, shaft wear has been practically eliminated when rubber bearings have been installed. Ordinarily their performance is best at high shaft velocities. The upper limit of speed at which rubber bearings can be used has not yet been determined. These bearings were used on "Miss England" which broke the world's record for motor-boats when a shaft speed of more than 12,000 r.p.m., equivalent to a peripheral speed of 4,330 ft. per min., was attained. At these higher speeds, one chief advantage of the rubber bearing is that it permits a shaft to rotate about its center of gyration even if this does not coincide with the exact geometrical center, thus reducing the vibration usually inherent in high speed shafts of this type.

There are of course certain limitations which control the use of rubber bearings. They are not ordinarily recommended for operation at temperatures in excess of 150 deg. F., although they are being successfully used on condensate pumps of commercial ships and naval vessels, where the temperature may at times reach 190 deg. F.

Oil and grease, especially mineral oil, even in small quantities are detrimental to rubber bearings as they soften the rubber and make it tacky. When in this condition the rubber is soon destroyed under the load of the rotating shaft.







Figs. 6 and 7—A comparison of the coefficient of friction and power loss respectively for rubber and metal bearings, when operated at 183 ft. per min. Fig. 8 shows the deflection in bearings of different diameters and lengths under various loads

QUESTION AND COMMENT

Cellulose Roll Feed

M. G. DEMOUGEOT
International Bending Machine Company

● Conventional idlers, brackets, guides and arms used to feed wrappers in cigar wrapping machines are difficult to apply to machines using cellulose wrappers. All parts of the cellulose feeder have to be adjustable. The diameter of the roll varies from 8 or 9 in. when full to 3 in., the diameter of the core. The weight of the roll varies from 5 to 6 lb. when full to about 2 oz. when empty.

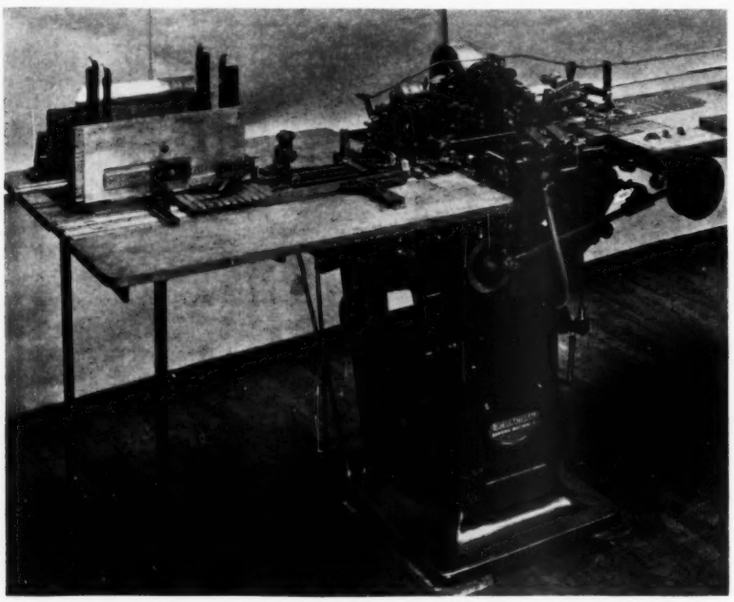


Fig. 1—Because of variations in size and weight of the roll of cellulose wrapper, high speed operation required a new type of feeding mechanism for this cigar wrapping machine

In the cigar wrapping and banding machine shown in Fig. 1, a new type of feeding mechanism makes it possible to apply cellulose wrappers to 4,000 cigars an hour. In this machine, cigars are placed in the rectangular hopper at the left, and are carried into the central part of the machine, one at a time, by a reciprocating conveyor. At this point a band and piece of cellulose is fed in position over the cigar. Plates and jaws fold the wrapper and band around the cigar, and electric heaters seal the folds as the cigar is carried out of the machine.

Rapid feed of the cellulose wrappers into the wrapping mechanism was accomplished by the device shown in Fig. 2. The roll of cellulose, shown at *A*, is placed on a 3-lb. core plug *B*. This added weight keeps the cellulose roll in contact with the supporting rollers *C* and *D* even when the last of the roll is being fed.

Cellulose strip drawn from the roll is passed around an aluminum roller *E*, and under the feed roller *D* where the strip is held in contact by pressure from the idler roller below. A measured length of wrapper is pulled from the roll, fed into the machine and cut off.

Control of the cellulose roll is obtained by means of the compound-arm mechanism pivoted at *H*, with a universal joint at *K*. Thus, the flat surface of the roll lays flat on the face of the feed rolls at all times.

As the cellulose roll becomes smaller, the arms *F* and *G* fold up. Any irregularities or wobbling of the core in the roll is taken up by the universal joint, and the two wings *L*, under the tension of *M*, a helical spring. Tension of this spring can be adjusted to prevent the roll from overrunning, eliminating slack in the wrapper feed.

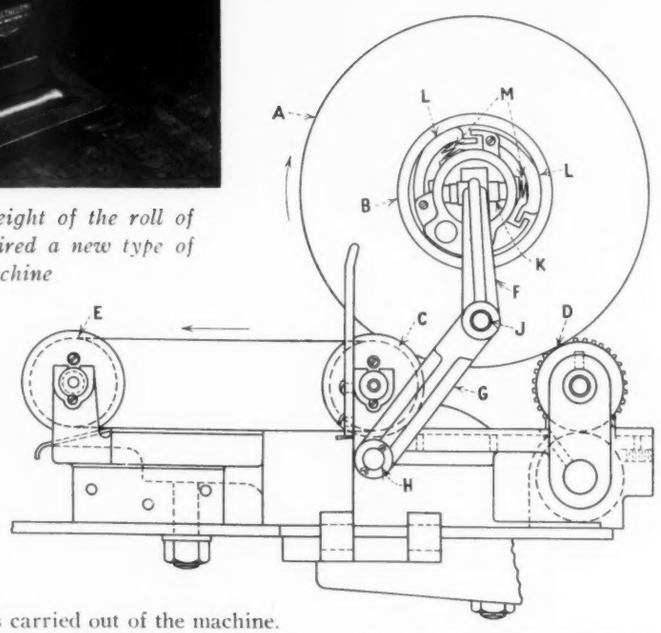


Fig. 2—The roll of cellulose rides on rollers, and is guided by jointed arms attached to a heavy core plug

Designating Drawings

To the Editor:

● No single system of drawing and pattern symbols can be applied to all branches of industrial activity. But whatever plan is used, it must indicate where the part belongs, and what subdivision it is of the complete whole. In one automobile engineering department, parts are identified as detail units of complete assembly units, and are also identified with the complete unit such as engine or transmission.

In this system the pattern and drawing number consists of one symbol (a letter) indicating the model, a second letter indicating all details belonging to the unit, and numbers indicating sub-assemblies and detail parts. Thus, the pattern number BA-1-1 designates a part for model B automobile, the part belonging to unit A, the engine, the numeral 1 designating the cylinder. Similarly BA-4-2 designates the flywheel detail for the same motor and model.

The control of sub-assemblies can be simplified by grouping all details on a special sheet so that one order number can include any number of units.

The stock sheet book should also be grouped to include all cast iron castings by drawing or pattern numbers. All bronze and aluminum castings should be listed in a similar manner, as well as other stock parts, such as sheet steel, brass tubing, rubber hose and fiber. —JAMES MCINTOSH

Cleveland Heights, Ohio.

Manufacturer's Liability for Injury Caused by Defects in Product

LESLIE CHILDS
Attorney and Counselor at Law

● The question of the liability of a manufacturer of a machine or other appliance for injury to a user from defects may depend both upon the nature of the machine and the territory or state in which the injury occurs. And, since the laws of the states are not uniform on this subject, it cannot be covered by any hard and fast rule.

Some states adhere to the rule that unless a machine is inherently dangerous, the manufacturer will not be liable for injury to a user from defects, in the absence of privity of contract. Other states hold that, if by reason of defects, a product cannot be used without reasonable certainty of injury, a manufacturer may be liable, even in the absence of privity of contract between him and the injured person.

As an illustration of how a manufacturer may be saddled with unexpected and substantial liability for injury caused by defects in his product, a Massachusetts manufacturer of coffee urns sold an urn to a caterer in Wisconsin in 1924. Seven years later, the buyer of the urn loaned it to a customer to be used at a social function. A guest at this function was asked to pour coffee, and while so doing the urn without warning toppled over, spilling hot coffee which resulted in severe burns.

The evidence tended to show that the urn collapsed because of a defect in soldering the wire bead to the base of the container. The flame from the spirit lamp furnished with it melted the solder which caused the container to fall.

Under the law of Massachusetts, the defendant's state, the defendant would not have been liable because the urn was not an inherently dangerous instrument, and there was no contractual relation between plaintiff and the defendant. But under the law of Wisconsin, where the accident occurred, defendant would be liable, even to third parties, if the defect was caused by his negligence and rendered the urn a probable source of injury to anyone using it.

In this situation, plaintiff sued the defendant in the Federal District Court of Massachusetts, and invoked the rule of lex loci (the law of the place). And as the Federal courts generally do, where there is no conflict with Federal law, the court applied to the facts of this case the law of Wisconsin, the law of the state where the accident occurred. Defendant was awarded a judgment of $4,000, and the court reasoned, in part, as follows:

"No positive testimony . . . was offered by the defendant as to the exact fusing point of the solder used in constructing this particular urn. Apparently, direct evidence as to the construction of any particular urn was not available in a factory making hundreds of such urns each year. . . .

"The manager of defendant's plant could not and did not undertake to say that some employee may not have used too little solder or solder of inferior grade in soldering the wire bead to the base of the container of this particular urn. The expert of the plaintiff expressed the opinion that the accident was due to a faulty design and construction in that there was no support for the urn except the thin film of solder by which its base was joined to the flat metal bead which alone rested on shoulders of the four legs; that if this solder melted, the urn would drop down and topple over.

"We think there was substantial evidence . . . that the plaintiff's injuries were due to the use in constructing this particular urn of solder of too low fusibility in joining the base of the urn to the metal bead; that the . . . evidence . . . tended to exclude any other explanation of the accident except the negligence of the defendant in designing and constructing this particular urn, for which the defendant is liable, since it resulted in an article reasonably certain to expose a third person to injury even when using it in the manner it was intended to be used. (72F. 2d 359)"

The foregoing case was carefully reasoned by the court, and the holding is in accord with the line of authorities that impose liability upon a manufacturer, for injury resulting from defects in his product that render the latter a probable source of injury when being used for the purpose intended. Under this rule, the machine, appliance, or article, does not have to be inherently dangerous, and the liability extends to users, irrespective of privity of contract.

The best form of insurance against being called upon to defend an action of this kind consists in the exercise of extreme care in design, construction, and inspection, where the product is one that may prove dangerous to the user if defects appear.

Strength of Steel

To the Editor:

● On page 33 of the January number, the item "Homogeneous Steel Ten Times Stronger," states that a study of molecular bonds would indicate that steel should have a tensile strength of about 1,000,000 lb. per sq.in. It is doubtful whether the failure of steel to attain this strength is entirely caused by lack of homogeneity. Large single crystals of iron that are certainly homogeneous show no marked increase

in strength over ordinary iron and steel, and in fact are not as strong as some steel wires.

The failure of steel to reach its full theoretical strength is caused by the existence of cleavage planes along which sliding readily takes place, and is also caused by the natural porosity of the metal. The strengthening effect of carbon and other additions to iron, as well as the effect of cold working, is to break up these cleavage planes, producing a non-homogeneous structure. Small diameter piano wire actually attains strengths up to 400,000 lb. per sq.in., with such a non-homogeneous structure.

Porosity in metals is established by the fact that they occlude large quantities of gas, and that metals cannot be used in the construction of high vacuum containers. In fact, other metals can actually be diffused into solid steel, penetrating rapidly to a depth of several inches. The accompanying increase in specific gravity indicates that the penetrating metal occupies space within the steel that was previously vacant.

Full theoretical strength of metal cannot be developed without both breaking up crystal planes and eliminating porosity. The resulting material would be extremely hard, and would no longer be recognized as a metal. However, attempts to strengthen metals should also take into consideration the effect of density.

—Leo G. Hall,
Allegan, Mich.

Graphical Determination of Principal Axes

HERMAN E. MAYROSE
University of Detroit

● When the rectangular moment of inertia of a section is known for one set of axes, the moment of inertia for any other set can be determined by the following graphical method. This construction is adapted from Land's circle, a method applied to problems of stress analysis, and has been given to students in applied mechanics for some time.

Using the section shown in Fig. 1 as an example, the moment of inertia about the axes X' and Y' are found as follows. Lay off line OA equal to I_x, and AB equal to I_y, where I_x and I_y are the known moments about the axes x and y. Draw AC normal to OB and equal to P_{xy}, the product of inertia with respect to the x and y axes. AC is laid off above the line OB if P_{xy} is a negative number, and below if positive. In the diagram the product is negative.

The product of inertia, P_{xy} is given by any modern text or handbook which contains a complete chapter on moment of inertia of areas.

Draw the circle with OB as a diameter, and draw OO' parallel to Y', the new y-axis. Through the center of the circle $O'D$ is drawn with the point C projected to C', making $C'C$ normal to $O'D$. For the new set of axes the distance $O'C'$, is equal to I_x, $C'D$ is equal to I_y and CC' is equal to P_{xy}, the new product of inertia.

When the product of inertia vanishes, the axes become the principal axes, and the moments of inertia are a maximum with respect to one axis and a minimum with respect to the other. The same construction, therefore, can be used to obtain the principal axes by locating the point C as explained before, and drawing $O'D$ through C and the center of the circle. The line OO' then will be parallel to the principal y-axis.

In applying this method to standard structural shapes, tables usually give I_x, I_y and the minimum moment of inertia. To find P_{xy}, the product of inertia, the construction shown in Fig. 2 can be used.

Lay off OB equal to I_x, AB equal to I_y and BE equal to I_{min}. Draw the circle with OB as a diameter and a concentric circle through E. Then draw AC normal to OB. The line AC is equal to the product of inertia. The sign will be negative or positive according to the location of the area with respect to the axes.

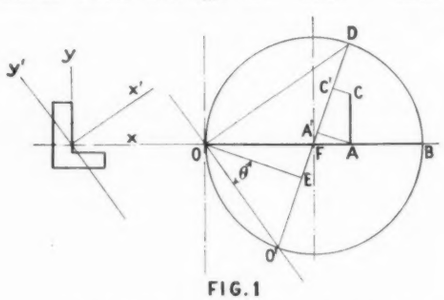


FIG. 1

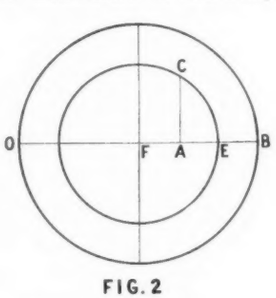


FIG. 2

Fig. 1—Using a modified Land's circle, moments of inertia for any axis can be found when the moments are given for one axis. Fig. 2—The principal axis can also be found by using the graphical method explained in the text

Built-In Lighting

To the Editor:

● Granting that two essential characteristics will determine the usability of a lighting system—namely quality and amount, or intensity—it is easily understood why it is often difficult or unduly expensive to secure both with a general broadcast, overhead lighting system. Often the seeing job involves the discrimination of color as in judging fire temperatures, or heated metal, or in the case of color blending, selections of copper versus brass, or noting stains on chromium or paint. This suggests special quality or special color illumination for specific close-up operations which need not be extended over an entire working area. Furthermore, many objects are viewed by contrast. In this case a relatively darker background may be needed or else a strong light from one single direction would help. For instance, the smoothness of a cut edge can best be judged by a grazing light coming from one side, rather than from a general overhead diffused light.

It is just being discovered that the human eye requires for easy seeing approximately the intensity of light that exists in the shade of a tree or building in daytime. This is usually more than 100 times the intensity that any general overhead lighting system provides. It therefore becomes necessary at the present costs of electricity to build up "hot spots" or high intensity localized lighting, especially with foolproof units built into the machine itself and movable with it, in order to even approximate daylight conditions.

Another requirement is that which calls for cleanliness of the lamp, re-

flector and accessory. The harder it is to reach the lighting unit the less attention it usually receives. Thus cleaning and good maintenance should be made easier in the case of well designed inbuilt machine lighting.

It is not unusual to find that certain operations on a machine require accurate seeing for only brief intervals. For example, when setting the gears on a lathe, or when centering a spindle, tying broken threads in weaving, or changing any adjustments as in the case of semi-automatic machinery, merely a momentary intense light is wanted at some particular spot. Otherwise, and for considerable periods of time, the machine may be in an illumination just necessary for safety or perhaps no more than five foot-candles. Such situations suggest providing a built-in unit giving an intensity of several hundred foot-candles. By controlling the circuit by a pushbutton the lamp will burn only when an operator is standing and viewing a particular setting or scale.

When machines stop working, as during lunch hours, cleaning periods, or at the end of the day, the built-in local lighting naturally is switched off and a cost savings is thereby effected.

—S. G. HIBBEN
Director of Applied Lighting
Westinghouse Lamp Company

Laying Out Optical Systems

To the Editor:

● As we had already made some studies on light concentration in connection with long-range light signals, we found considerable interest in the article, "How to Lay Out Optical Systems," by R. C. Hitchcock. However, we cannot agree with the author's nomogram using the last three scales to the right, as shown on page 451 of the December number. A more logical definition for the efficiency of filament to beam is the ratio of the acceptance angle to the entire radiated light flux.

As shown in Fig. 1, the acceptance zone *BKC* for the parabolic mirror is much larger than the zone *BAC* for a lens, showing a much higher efficiency of filament to beam for the reflector. The higher efficiency is attested by the fact that all auto headlights are made of parabolic mirrors rather than lenses.

The author's diagrams for lens and mirror under his nomogram are confusing, because they attempt to combine acceptance angle with magnification. Following the same designations used by the author, magnification M in a lens system is,

$$M = S/s = D/d$$

But the relation between the projection distance D and the image distance

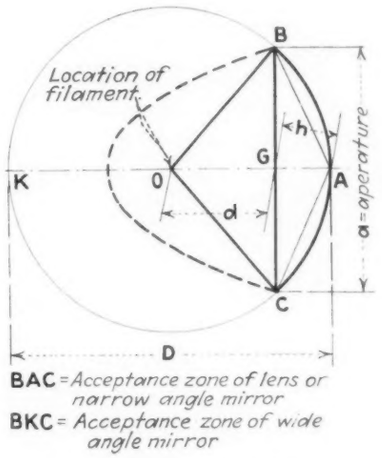


Fig. 1—Efficiency of either reflector or lens system depends on acceptance angle rather than on magnification

d is fixed by the focal distance f, and is given by,

$$1/f = 1/D + 1/d$$

The principal focus in the lens system shown in Fig. 2 is the point p to which parallel rays are brought to a point or focus. The optical center O of a lens or system of lenses is the point such that a ray passing through it emerges parallel. A point in the image is formed by the intersection of two rays from the filament, as shown in Fig. 2. One ray, parallel to the axis, is refracted through the principal focus, and the other ray passes through the optical center.

Several graphical methods of calculating the relation between f, D and d are available, but for most designs the relations are given quickly by the following simple construction. Assuming f and D given, draw perpendiculars to a base line at any convenient distance apart, with the height of the perpendiculars proportional to the given values of f and D. Draw an inclined line from the upper end of D through the upper end of f, extending the line until it intersects the base line. Erect another perpendicular to the base line at this point. Draw another inclined line from the base of D through the upper end of f, extending the line until it intersects the third perpendicular. This intersection defines the length d along the last perpendicular line.

In a parabolic reflector all rays emanating from the focus are reflected parallel; rays from any other point will be scattered and lost in dispersion. But no reflector will focus all rays coming from any assumed point in the object to the same point in the image; all reflectors are subject to some aberration.

If the filament is small relative to the focal length, then the small arc of the parabola utilized does not differ essentially from that of a sphere with a radius 2 f. Then, it can be shown that for a reflector, the focal distance in terms of projection distance and image distance is given by the same relation shown above for the sphere. Likewise, magnification for the reflector follows the same law as given above for the lens system.

—CARL P. NACHOD
Louisville, Kentucky

To the Editor:

● The nomogram given in the December number, page 451, was designed only for use with optical projection systems having relatively small apertures. The analysis given by Mr. Nachod is more accurate than the chart for systems having large apertures.

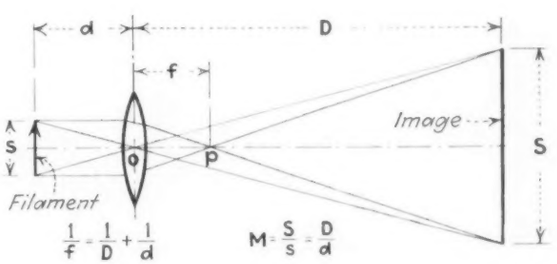


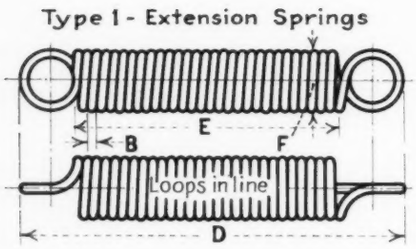
Fig. 2—Filament distance for a given image distance depends on f, the focal distance of the lens

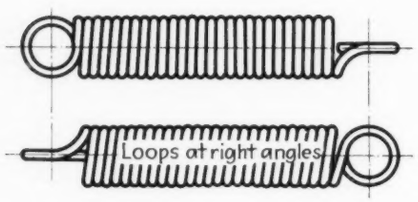
In many applications for small optical projectors an exact layout is not necessary. If the optical system is provided with a sufficient adjusting range, a number of these design factors can be taken into account in the final focusing or adjusting of the reflector. For long-range or high-power projectors, a more accurate design procedure as suggested by Mr. Nachod would be desirable.

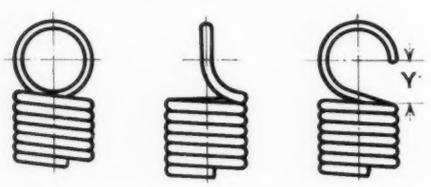
—R. C. HITCHCOCK
Westinghouse Electric & Mfg. Co.

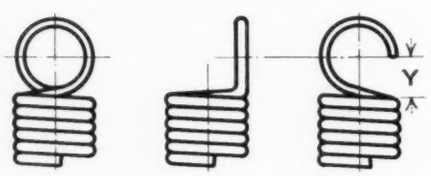
Spring Data

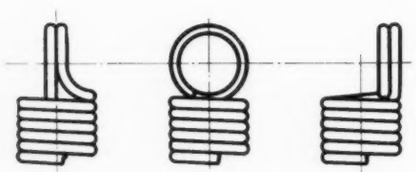
M. G. DEMOUGEOT
Assistant Factory Manager
International Banding Machine Company

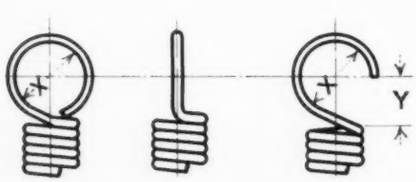


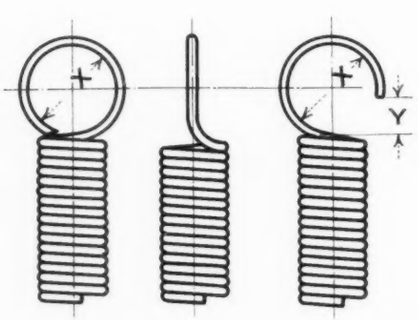


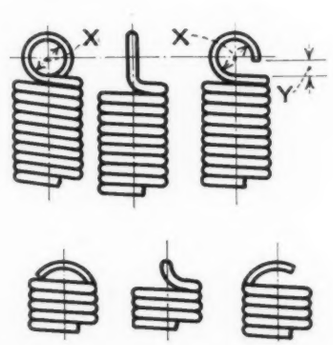












● SPECIFICATIONS REQUIRED for various types of springs. Type 1 for extension springs; Type 2 for compression springs; Type 3 for torsion springs; Type 4 for flat springs.

Types 1-2-3-4
A—Kind of material—music wire, chrome vanadium, phosphor bronze, etc.

Types 1-2-3-4
B—Size of material—round, square or flat.

Types 1-2-3
C—Style of spring—extension, compression or torsion.

Types 1-2-3
D—Length of spring overall.

Types 1-2
E—Length of spring in body.

Types 1-3
F—Outside or inside diameter.

Types 1-2
G—Style of ends—closed or open loops (plain, squared, ground, or squared and ground.)

Type 1
H—Distance to be extended.

Types 1-2-3-4
J—Load to be applied.

Type 1
K—Dimensions *X* and *Y* for special loops.

Type 1
L—Dimension *L* for swivel loops. (Usually made one gage size heavier than wire in spring.)

Type 2
M—Outside or inside diam. with plus or minus tolerance, depending whether spring works on rod or in a hole.

Types 2-3
N—Pitch or number of full coils.

Type 2
P—Distance spring will be compressed.

Type 2
R—Wound right or left hand.

Type 3
S—Size of rod, if any, on which spring is to work.

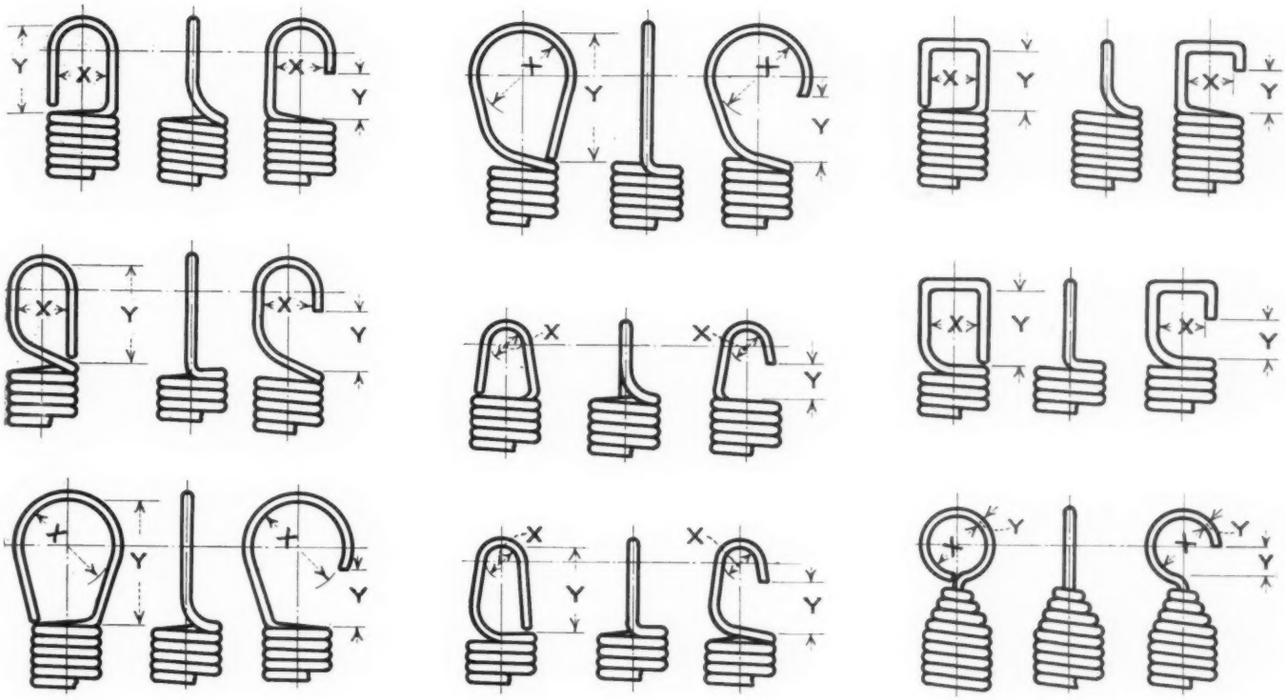
Type 3
T—Length and position of spring ends.

Type 3
V—Distance which spring ends are to travel.

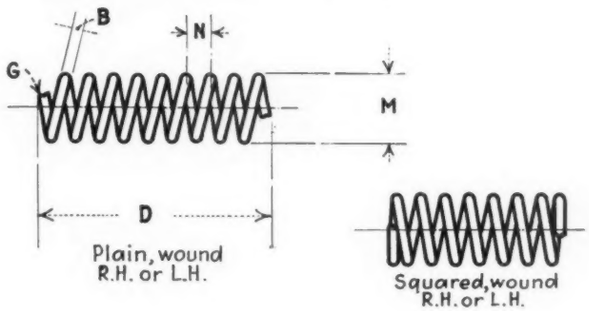
Type 4
W—Kind of material and temper.

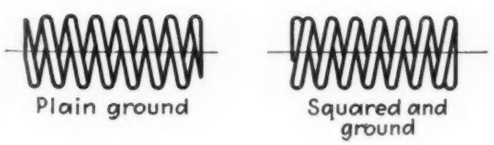
Type 4
X—Style of flat spring—flat-stamped, coiled or formed.

Type 4
Y—Distance spring will have to flex.

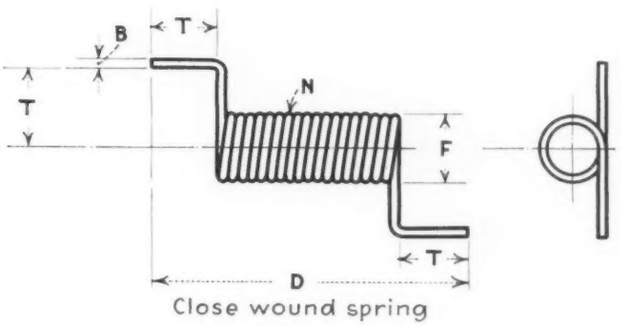


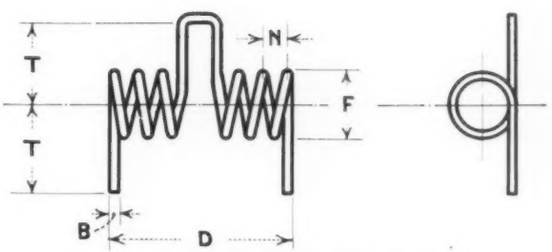
Type 2-Compression Springs



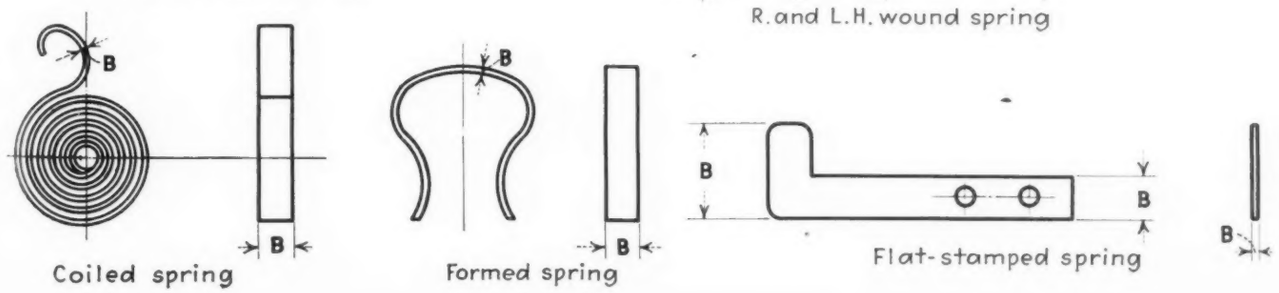


Type 3-Torsion Springs





Type 4-Flat Springs



NEWS

Ceramic Research At M.I.T.

THE notable contribution of ceramics to industry and opportunities for applying advanced knowledge in physics and chemistry to future developments have led the Massachusetts Institute of Technology to establish advanced degrees in this field beginning immediately. An investigation of the origin of clays is to be conducted under the direction of Prof. Frederick H. Norton of the department of mining and metallurgy, as the result of a recent grant by the Geological Society of America. The ceramics course emphasizes creative work rather than the study of manufacturing details. Candidates for the doctorate must have, in addition to strictly ceramic subjects, a thorough knowledge of physics, chemistry, combustion, furnace design, x-ray and crystal structure, drying, heat measurements and colloids. Among the advanced special studies offered are optical ceramics, product investigation, fundamental processes, properties, testing and special problems.

During the past 20 years the Institute's course in ceramics has led to many original contributions to the field of product engineering, including asbestos shingles, kaolin brick, asbestos board, insulating refractories and machines for molding silica brick. Through laboratory research much light has been thrown on the nature of plastic phenomena and the mechanism of color in the form of new glazes and materials. Studies are now being made of quick methods of firing and forming, fresh decorative processes and the adaption of new materials to ceramic purposes.

Foundrymen Plan Joint Meeting

THE Chicago section of American Foundrymen's Association, the Western Society of Engineers and the Chicago section of the American Society of Mechanical Engineers are sponsoring a joint meeting on "Engineering Uses of Modern Cast Metals," to be held March 18. The meeting will be held in the auditorium of the Engineering Building, 205 West Wacker Drive, Chicago, Ill.

Three outstanding men, one each from the cast steel, malleable iron and gray iron divisions of the foundry industry, will discuss the recent advancements that have been made in the engineering properties and applications of those three metals.

Advances in gray iron will be discussed by H. Bornstein, chief chemist and metallurgist, Deere & Company, recognized as an international authority on that subject. Progress in malleable iron will be discussed by D. P. Forbes, president and general manager, Gunite Foundries Corporation. Developments in the cast steel field will be told by A. N. Connarroe, metallurgist, National Malleable & Steel Castings Company. The chairman of the meeting will be announced later.

An exhibit showing the developments in foundry technique, physical properties and special applications of cast ferrous metals of special interest to designing engineers, will be held in conjunction with the meeting.

Electric Appliance Sales Increase

IN speaking to a group of 50 business press editors, A. E. Allen, vice-president in charge of merchandising of the Westinghouse Electric & Manufacturing Company, predicted an upward trend in business, particularly in the field of electrical household appliances. According to Mr. Allen, who quoted the results of surveys made by leading manufacturers of electric refrigerators, plans are being made for the largest year the industry has ever enjoyed.

Household refrigeration has grown every year since its beginning and only in the year 1932 have sales fallen below those of the previous years. Over the period from 1921 to 1934 the number of units sold has increased from 5,000 to 1,350,000, and the industry expects that more than 1,500,000 electric household refrigerators will be purchased in the coming year.

Similar optimistic predictions were made with reference to other electric appliances including dishwashers, clothes washers, irons, fans, vacuum sweepers and food mixers. Business in some of these lines increased in 1934 as much as 106 per cent over the 1933 figures.

One striking development has been that of the electric range business, which is looked upon as a barometer for sales of all household appliances. During the year 1934 the sale of electric ranges by Westinghouse was more than double that of 1933.

According to Mr. Allen, there are 20 million wired homes, representing about 30 per cent saturation. There are slightly in excess of 6 million refrigerators now installed in homes, many of which are now obsolete or worn out. Speaking about the design of refrigerators, both those of Westinghouse and those of other manufacturers, Mr. Allen commented that in some instances the cabinets are now of all-steel construction, without any wood; compressor units have been made more quiet and trouble-free and hermetically sealed in some instances.

A.S.T.M. Features Paint Symposium

A SYMPOSIUM on paint will be the technical feature of the 1935 A.S.T.M. Regional Meeting, to be held in Philadelphia, March 6, at the Warwick Hotel. Fourteen formal papers devoted to the symposium will be presented at the two sessions, morning and afternoon. Recognized technologists will present the papers. While at recent A.S.T.M. meetings there have been formal discussions on specific topics related to paints, such as hiding power, pigments, etc., the subject of preservative coatings was not covered from a broad viewpoint.

The symposium has been planned to deal with the various related subjects in a broad, comprehensive way; to present information about paint and paint materials that will be of particular interest to engineers and others using the various products. The papers should be of interest to users in many fields and not solely to paint technologists.

The engineers and chemists who will participate in the program bring to it a wide experience in many branches of the industry and an interesting meeting is anticipated. The program includes the following papers: "Preparation, Use and Abuse of Specifications for Paint Materials," P. H. Walker, assistant chief, Chem-

istry Division, National Bureau of Standards; "Protective and Decorative Coatings for Railroads," A. M. Johnson, engineer of tests and chemist, The Pullman Company; "Paint Testing," C. D. Holley, director of paint research, Acme White Lead and Color Works; "Varnish Testing," W. R. Fuller, technical director, Pratt & Lambert, Inc.; "Lacquer and Lacquer Testing," H. E. Eastlack, director, Parlin Laboratory, E. I. duPont de Nemours & Company; "Chemical Colors," A. F. Brown, general manager, Imperial Color Works; "Natural and Synthetic Resins," W. T. Pierce, The Resinous Products & Chemical Company; "Solvents and Volatile Thinners," R. M. Carter, research chemist, U. S. Industrial Alcohol Company.

This meeting will be open to all who are concerned with paint and paint materials, their production, application or testing.

Leading Concerns Enter Art Exhibit

COOPERATION from leading concerns representing more than 20 separate industrial fields has been pledged to the Industrial Arts Exposition of 1935, according to an announcement by the National Alliance of Art and Industry. Already, contracts for space have been signed by 26 exhibitors, all of them outstanding in their special fields.

Visitors to the exposition, to be held at Rockefeller Forum, New York City, April 15 to May 15, will discover the latest product for their service and comfort from every important sector of industry. Products of the silversmith and the foundry, the loom and the glass blower, will be shown, as well as the more intricate devices of the electrical manufacturer, clockmaker and the maker of weighing machines.

The exhibits will be shown under the groupings of housing, household appliances, transportation, communication, fabrication and leisure, each group to be shown in relation to an ideal presented by a leading designer, decorator or engineer.

Among the firms which have already engaged space at the Industrial Arts Exposition, which will occupy all of the 44,000 square feet of Rockefeller Center Forum, are the following: American Car & Foundry Company; American Cyanamid Company; Bakelite Corporation; Breskin & Charlton Publishing Company; Doehler Die Casting Company; General Alloys Company; International Nickel Company; International Silver Company; Raymond Loewy; Montgomery, Ward & Company; National Retail Dry Goods Association; Ben Nash; Owen Illinois Glass Company; Philco Radio Corporation; R.C.A.-Victor Corporation; Sears, Roebuck and Company; Eugene Shayne; The Shelton Looms; Sherwin-William Company; Sight Light Corporation; Tennessee Eastman; Toledo Scales Company; Toledo Synthetic Products, Inc.; Walter Teague; Underwood & Underwood; Van Doren & Rideout; Western Clock Company.

Ben Nash is chairman of the exhibition committee of the National Alliance of Art and Industry, and T. J. Maloney, 386 Fourth Avenue, New York, N. Y., is manager of the exhibition.

Mechanical Properties of K Monel

Hardness Grade	In thousands lb. per sq. in.			Min. Elong. in 2 in.	Min. Per Cent Red. in Area	Brinell Hardness 3,000kg. Load
	Tensile Strength	Yield Point at 5% Elong.	Propor. Limit			
A	120 max.	80 max.	60 max.	40	50	softer than 225
B	120 min. 140 max.	80 min. 100 max.	60 min. 80 max.	30	35	225–275
C	140 min. 160 max.	100 min. 120 max.	80 min. 100 max.	20	25	275–325
D	160 min.	120 min.	100 min.	15	20	greater than 325

Visit Leipzig Fair

KENNETH H. CONDIT, editor of *American Machinist* and *Product Engineering*, sailed on February 23 to attend the Leipzig Trade Fair and to study the 2,500 exhibits in the engineering section. After the Fair he will spend several weeks visiting plants in Germany and England.

Marc Rose, editor of *Business Week* will accompany Mr. Condit to the Leipzig Fair and will then make a short trip covering Austria, Italy and France. Both editors will report on the latest European developments in the fields of their respective publications.

MEETINGS

American Society for Testing Materials—Annual meeting, Book-Cadillac Hotel, Detroit, Mich., June 24-28. R. E. Hess, assistant secretary, 260 South Broad St., Philadelphia, Pa.

American Society for Testing Materials—Paint Symposium, Warwick Hotel, Philadelphia, Pa., March 6. R. E. Hess, assistant secretary, 260 South Broad St., Philadelphia, Pa.

EXHIBITIONS

Industrial Arts Exposition—Rockefeller Forum, 30 Rockefeller Plaza, New York, N. Y., April 15-May 15. T. J. Maloney, manager, 386 Fourth Ave., New York, N. Y.

New Aluminum-Monel Alloy Announced

AT THE annual meeting of the American Institute of Mining and Metallurgical Engineers, Dr. W. A. Mudge, metallurgist, and Dr. Paul D. Merica, director of reserach, both of the International Nickel Company, announced a new Monel alloy developed in the laboratories of their company.

Called K Monel, this alloy contains about 4 per cent aluminum and is claimed to possess remarkable corrosion and heat-resistance properties in combination with high strength.

By variations in heat-treatment, cold-working and aluminum content, K Monel can be produced to give a wide range of mechanical properties. it is produced only in the rolled form and in four hardness grades, *A*, *B*, *C* and *D*. The mechanical properties for these four grades are given in the accompanying table.

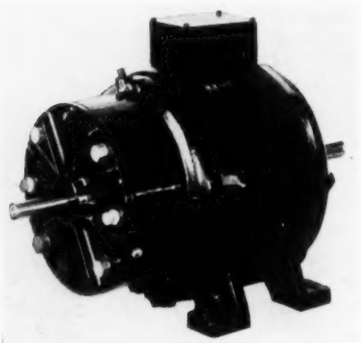
Hardness grade *A* is the softest, the softness being obtained by a softening heat-treatment. Grades *B* and *C* are of an intermediate hardness obtained by heat-treatment. Grade *C* is the hardest that can be machined. Hardness *D* is produced by cold-working and heat-treatment.

Among the uses for which this new alloy is particularly suited are turbine blades and pump rods, for which applications its corrosion and heat-resisting properties in combination with its high strength, will be valuable.

New Materials and Parts

Motor With Magnetic Brake

A disk type brake, built integrally with the motor, which holds the load by spring pressure whenever the power is shut off from the motor. The electro-magnet windings are connected with the motor supply leads



inside the motor, which is said to simplify installation and assure correct wiring. Brakes are furnished built into a.c. and d.c. motors of single, multiple and variable speeds, and are used for hoists, winches, cranes and similar applications. Electric Specialty Co., Stamford, Conn.

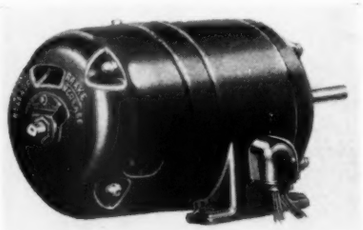
Speed Reducers—Horizontal

Type MA shown at the left has a single reduction right angle worm drive. The steel worm is hardened and ground and the worm wheel made of phosphor bronze. Both motors are mounted on anti-friction bearings and have oil seals to prevent leakage. Type MA has torque capacities up to 263 lb.-in., and is said to run continuously at its rated capacity. Available in ratios of $3\frac{3}{4}$ to 1 up to 48 to 1, and with $\frac{1}{6}$, $\frac{1}{4}$ or $\frac{1}{3}$ hp., a.c. or d.c. motors. Can be furnished without motor.

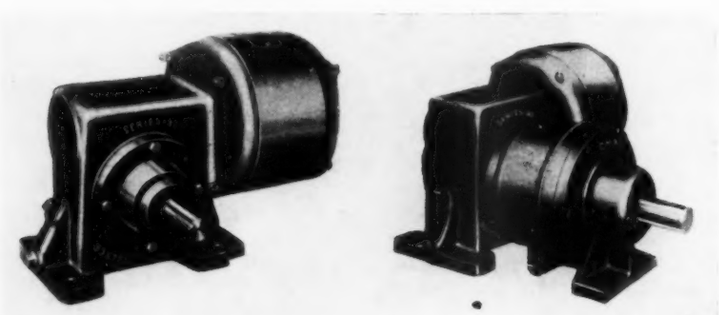
Type CMA at the right has a compound reduction right angle gear drive. The first reduction is by worm and wheel and the second reduction through a series of spur gears. The shaft may extend either side for two widely different speeds. Has torque capacities up to 1,500 lb.-in. Available in ratios up to 5,000 to 1 with $\frac{1}{6}$ hp. a.c or d.c. motor. Boston Gear Works, Inc., North Quincy, Mass.

Multiple-Disc-Brake-Motor

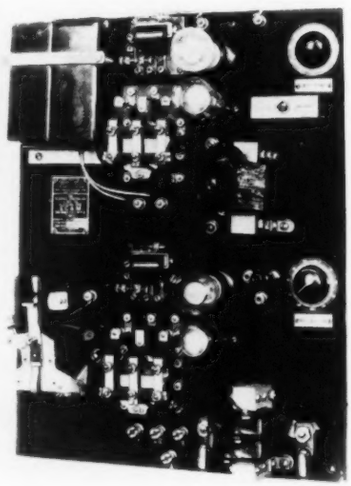
The brake has a spring set and electro-magnetic release and is housed in a special end cover. The braking disk revolving with the rotor is of hardened and ground steel, and is carried on a splined heat-treated alloy steel shaft extension. Brake is adjustable without removing cover. A 1,725



r.p.m. motor is said to stop within 5 or 6 revolutions when brake is applied. Available for single voltage, continuous or intermittent duty and in 9 sizes from 4 to 850 ft. lb. maximum retarding torque. Master Electric Co., Dayton, Ohio.



Automatic Repeat Weld Timer



Similar to their standard timer, with an additional timing circuit for governing the length of time that the electrodes of the welding machine are separated. Two independent and adjustable electrical circuits are provided so that time interval can be adjusted to suit operator's needs. The Electric Controller & Mfg. Co., 2702 East 79th St., Cleveland, Ohio.

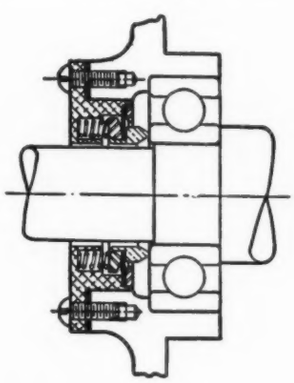
Hydraulic Power Unit

Used for drilling, reaming, counterboring, spot facing and similar operations. Adjustable for length of stroke, rapid traverse and feed. Feed rate can be changed from outside by graduated adjusting valve while unit is feeding, permitting selection of the most suitable feed. There are two forward feeds each independent of the other. Adjustable dogs for controlling the rapid traverse, feed, stop and reverse of the unit, are provided. Hardened and ground alloy steel has been used wherever possible. An electric motor is mounted on an adjustable base located above the unit and drives the hydraulic pump and driveshaft through V-belts. The pump driving shaft is connected through change gears to the spindle drive shaft. The spindle is supported front and rear in the main quill which carries a piston that is hydraulically operated. By changing pressure at

either end of the piston, the main quill or ram moves either forward or backward providing the feed for the unit. The feed of the spindle can be altered by removing the cover plate at the rear of the unit and changing the feed gear. The quill is furnished with a mounting flange for attaching multiple spindle heads. Sight gages are provided to show oil level and pressure of the hydraulic unit, the oil pressure being adjustable from the outside of the unit. Ex-Cell-O Aircraft & Tool Corp., 1205 Oakman Boulevard, Detroit, Mich.

Oil Seals for High Speed

A heat resisting fabric diaphragm type of seal for use with ball, roller or plain bearings, protects the bearing against entrance of foreign material and loss of lubricant. The lapped bronze ring which bears against the shaft shoulder or the inner race of anti-friction bearing, through a series of compression springs, has up to $\frac{3}{16}$ in. adjustments, depending on size, to take up wear and play of shaft, automatically. To avoid torsional strain on the diaphragm all sizes above 1 in. diam. have guide pins, sliding in wells in the housing. Available with flange for end and outside mountings, or without flange for internal and press fit mountings. Both illustrations show the flanged type of seal. Sizes of $\frac{1}{4}$ to 4 in. by $\frac{1}{16}$ in. can be furnished in each style. Gits Bros. Co., 1859 S. Kilbourn Ave., Chicago, Ill.



Safety Switch

In sizes from 30 to 600 amp. capacity. Equipped with interlocking door which prevents operation of switch with door open or opening door with switch closed. Semi-floating, double break V-blades and stationary contacts backed with springs, said to guarantee full contact under heavy pressure. Inclosed in narrow rust-proof steel case. Sponge rubber sealing gasket between door and cabinet is said to make switch weather-proof and dust-tight. The Electric Controller & Mfg. Co., 2700 E. 79th St., Cleveland, Ohio.

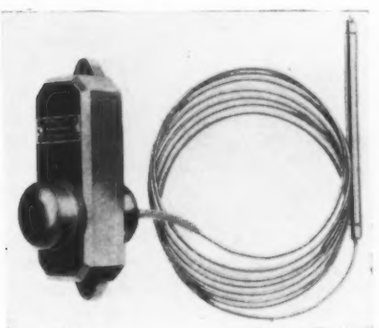
Motorized Speed Reducer

Is used for operating valves, dampers, filters, automatic controls, moving displays, and small machines that must be driven at slow speeds. One housing is used for all sizes of motors and gear ratios. Countershaft is parallel to motorshaft which makes it easy to mount on the driven apparatus. Spur gears are used which, the quietness, are driven by a composition gear on the first gear cluster. Ball bearings are used for motor end bearing, and bronze sleeve bearing for reducer shaft. This reducer may be operated in either vertical or horizontal position. Motors

are available for split-phase a. c., shunt or compound wound d. c., series wound d. c., and three-wire reversible type motors for continuous duty service. Horsepower ratings 1/50 hp., at 1140 r.p.m., 1/20 hp. at 1725 r.p.m. and 1/20 hp. at 3450 r.p.m and gear ratios range from 12 to 1 up to over 6,000 to 1. Janette Mfg. Co., 556-558 W. Monroe St., Chicago, Ill.

Differential Thermostat

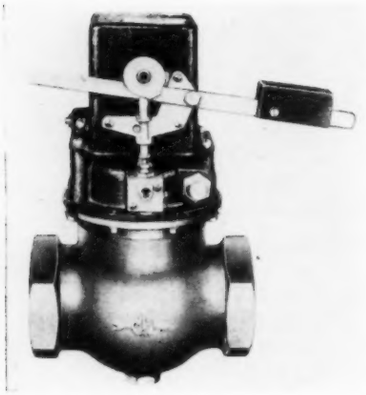
Designed primarily for temperature control with room cooling apparatus. Operates on principle of two vapor charged thermal elements opposed to each other, one actuated by room tem-



perature, the other from outside. Said to be accurate to approximately 1 deg. F. Available up to 25 watts, at 25 volts a.c. or d.c. current. Detroit Lubricator Co., Detroit, Mich.

Gas Valve—Slow Opening

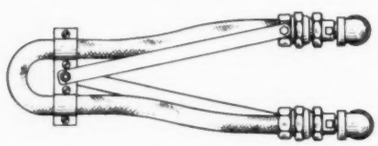
The opening time is approximately 30 sec. and closing time approximately 3 sec. Motor opens valve while a 50 lb. spring rotates motor backwards to close the valve. Adapter includes an adjustable flare pilot with by-pass adjustment for on-off or low-high-low service. The damper lever arm and the flare pilot are operated from a cam on the motor crankshaft. Provision is made for operating valves manually during power failure. An automatic recycling feature returns valve to command of controller when current service is restored. Valve sizes $1\frac{1}{2}$, 2, $2\frac{1}{2}$ and 3 in. are straight-



through screwed type with bronze bodies. The 3, 4 and 6 in. sizes are straight-through, flanged type, with iron bodies. The valve carries a 3 lb. pressure rating up to and including 3 in. size and 1 pound rating for 4 to 6 in. sizes. Minneapolis-Honeywell Regulator Co., Minneapolis, Minn.

Flexible Tube With Bracket Support

Flexible tubing is seamless and free from welds, joints, seams, laps or packing, and is made from drawn tubing of special bronze. Deep helical corrugations, die-formed into the tubing impart necessary flexibility. Bracket is attached to flexible tubing in such a manner that flexing of tubing is equal at all points, and tubing is held constantly in a horizontal position, thus preventing sagging and possibility of liquid pockets forming. Applicable either in series or in individual mountings in a number of sizes for various movements. The American Metal Hose Branch of the American Brass Co., Waterbury, Conn.



Electrical Indicating Instrument

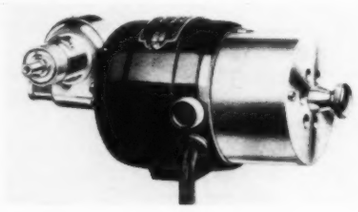
In a square case for switchboard mounting. Available in types for direct current, radio frequency and rectox types for measuring alternating current. Only one large hole in the panel is required for mounting and all visible mounting screws are eliminated by using mounting clamps. Soldering clips are standard for electrical terminals of these instruments. They have a scale length of 2.4 in. with accuracy of 2 per cent (rectox type 5 per cent). Flange dimensions are 3x3⅛ in. The permanent magnets are made of tungsten or cobalt steel, heat treated and aged. The soft iron pole tips are accurately aligned and fastened to the permanent magnet by electric welding, assuring perfect alignment and uniform air gap for the moving coil. The iron core is accurately aligned by a die cast bracket which supports sapphire bearings and springs. Standard dials are finished in matte silver with black figures. All instruments are furnished with a solder terminal. A notch is provided to wrap the wire around the terminal. These solder clips project straight out from the side of the base. Westinghouse Electric & Mfg Co., East Pittsburgh, Pa.

Oil-Tight Solenoid Switches

Type H for hazardous gas locations, is encased in a cadmium-plated cast-iron hood and oil tank, which is machined to maintain a tight seal. The hood is capped to receive conduits which can be threaded in to make a gas-tight seal. Used in chemical plants where both inflammable and corrosive gases or vapors are present. Type J starter is used where oil immersion is required only as a protection against corrosive gases, but is not intended to meet explosion-proof requirements. This type is inclosed in a sheet metal oil tank and has a cast-iron hood both of which are cadmium-plated. Conduit can be threaded into the top. The double-break silver alloy contacts require no maintenance and eliminate flexible connectors. Soldered-type thermal overload relays which can be reset without opening the cabinet provide protection against overload. Available for polyphase motor requirements up to 15 hp., 110 volt; 30 hp., 220 volt, and 50 hp., 440-550 volt, also self-starting single phase motors. Allen-Bradley Co., 1311 S. First St., Milwaukee, Wis.

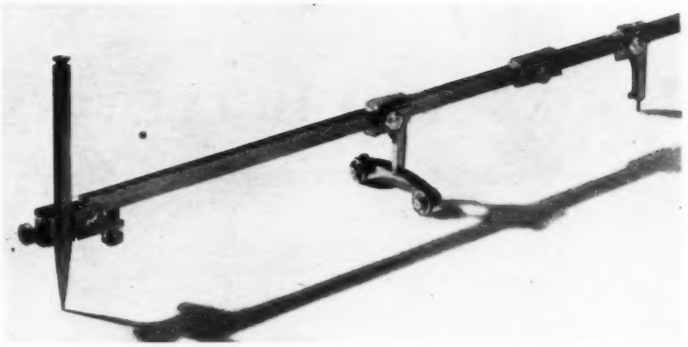
Governor Controlled Series Motor

The advantage of constant speed is combined with the light weight and power of series universal motors, in which two types of governors are offered, the adjustable type on which the speed can be varied while the motor is in operation and the fixed type which can be adjusted for a given speed only when the motor is at a standstill. Governor serves to hold motor speed constant, regardless of minor fluctuation in line voltage, and is adapted to apparatus where quick acceleration is required, yet where constant speed under varying load is necessary. Three sizes each of the two types are available for controlling the speed of motors rated at 1/64 to 1/8 h.p. over a range of approximately 1,000 to 7,500 r.p.m. The Dumore Co., Racine, Wis.



Beam Compass

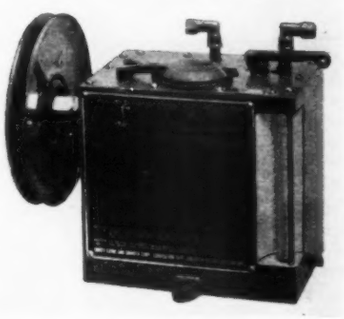
A new style beam compass that will hold any ruling, pen, pencil or other square, round or odd shaped marking tool. The main unit has a 10 in. rectangular beam which may be fitted with a 15 in. extension. A micrometer adjustment, a clamp screw, a four inch scriber and a sliding leg complete the main unit. A wheel attach-



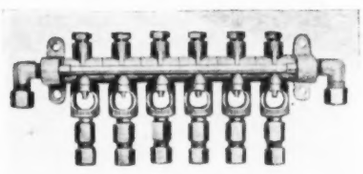
ment used for dotting lines and drawing large circles is available. The scriber and needle leg may be fitted with caliper points if desired. This instrument is made of duralumium for lightness, and all parts have a satin finish. The Wade Instrument Co., 2274 Brooklyn Station, Cleveland, Ohio.

Pulsating Lubricator

An automatic oil lubrication syster for industrial machinery. Is driven by a flat or V-belt, chain or gearing from a $\frac{1}{8}$ hp., 1,750 r.p.m. motor, or direct-connected through a shaft coupling. The pumping unit is said to maintain a constant low pressure circulation of oil through loop line with periodic quick impulses of high pressure to the feeders. The frequency of these pulsations range



from 8 per min. to 1 every 6 min., as determined by the pumping unit. One pump supplies up to 100 feeders, and the loop line may be as long as 100 ft. The pump is of the piston type with positive valves. The reservoir has capacities from 3 qt. to 4 gal. Double strainers prevent dirt entering the system and an oil level gage is provided. The feeder valves are



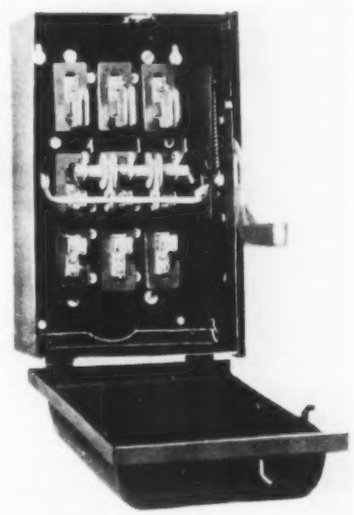
actuated by the pump pulsation. Upward flow of oil through the seat prevents clogging. These feeders may be arranged separately or in gangs of from 2 to 20 or more. Three styles are available. Style M is adjustable and has a sight glass. Style F is of the rated feed, non-adjustable type without sight glass, and is used for individual feeds of oil upwards, downward or horizontally. Style MA is non-adjustable and has a definite feed and shut off, with sight glass. Rivett Lathe & Grinder Corp., Brighton, Boston, Mass.

Air Hose

Compensated type, in which a braided cord member for longitudinal stresses and a spiral member for expansion in lateral strength are welded in age-resisting rubber. Said to compensate the usual unequal stresses, eliminate internal friction and resist surge and pulsation. The Manhattan Rubber Mfg. Div. of Raybestos-Manhattan, Inc., Passaic, N. J.

Square D Switch

Has an elevated removable base for simplified wiring. Embodies a front operating handle, downward opening cover and visible blades. Said to take up less space than former design. Modernistic cover has bevelled edges. Obtainable in 30 amp. 2 and 3-pole fused and unfused, and 3 and 4-wire solid neutral. Square D Co., Detroit, Mich.



MANUFACTURERS' PUBLICATIONS

Automatic Lubricators—Rivett Lathe & Grinder Corp., Brighton, Boston, Mass. Bulletin B-5, 16 pages, 8½x11 in. Describes "Blanchard Pulsator" and oil feeders with application photographs and cross-section line cuts. Installation and dimension charts included.

Carboloy—Carboloy Co., Inc., Detroit, Mich. Booklet SC-35, 36 pages, 8½x11 in., complete information on the "Profitable Use of Carboloy Tools," illustrates numerous types of tools, also materials machined with cemented carbide. Tells how tools are made and gives physical characteristics.

Cold-Drawn Steel—Union Drawn Steel Co., Massillon, Ohio. Folder describing physical properties of SAE 1020 steel bars before and after drawing, and listing products made by this and other grades of steel.

File & Tool Handles—J. L. Osgood Handle Co., 43-45 S. Pearl St., Buffalo, N. Y. Pamphlet, 10 pages, 4x9 in. Describing a new line of balance grip and plain type indestructible handles for files, soldering irons, screw drivers and wood turning lathe tools, listing sizes and prices.

Industrial Rubber Goods—The B. F. Goodrich Co., Akron, Ohio. Engineering data book, 23 pages, 8½x11 in., describing more than 200 rubber items, among them belting, hose and packing. Contains design data and tables for belts and discusses the relative merits of rubber belts and other materials.

Oil Seals for High Speed—Gits Bros. Mfg. Co., 1859 S. Kilbourn Ave., Chicago, Ill. "Special Metal Diaphragm Seals." Folder, 4 pages, 8½x11 in., illustrating non-rotating seals with and

without flanges, which seal the oil in the housing by use of a spring backed bronze ring bearing against shaft shoulder in a plain bearing or the inner race of an anti-friction bearing. A list of sizes from ¼ to 4 in. advancing by ⅛ in. is included.

Pyrometer — C. J. Tagliabue Mfg. Co., Park and Nostrand Aves., Brooklyn, N. Y. Bulletin No. 1101, 8½x11 in. Describing photo-electrically balanced recorders, photoelectric controllers, light beam indicators and potentiometers for thermocouples and accessories. Photographs show inside mechanism.

Resistance Charts—Hardwick, Hindle, Inc., 40 Hermon St., Newark, N. J. Four 8½x11 in. self-calculating charts for simplified determination of relation between resistance, voltage, current and power of any resistor or rheostat.

Roller Bearings—McGill Manufacturing Co., Valparaiso, Ind. Bulletin No. 35 showing various kinds of full type "Multirol" bearings having both needle or larger rollers. Illustrated with photographs, line cuts of application and has several pages of design data.

Shafting—Bliss & Laughlin, Buffalo, N. Y. Folder, 4 pages, 8½x11 in., giving data on turned, drawn, ground and polished shafting with sizes, tolerances and applications.

Stainless Steel—The American Rolling Mill Co., Middletown, Ohio. Folder, 4 pages, 8½x11 in., "Fine finishes make Fine Products," describes characteristics of Armco 17 and 18-8 stainless steels, giving composition and facts of these two grades and their uses.

"Structured Steel"—LaSalle Steel Co., Palmolive Bldg., Chicago, Ill. Booklet, 8 pages, 9¾x12 in., giving history of steel specifications and discussing structures and chemical analyses. Photomicrographs illustrate the possible range of structural variations in carbon steels with identical chemical compositions and emphasize the importance of the control exercised in the making, rolling and drawing of the steel.

Washers—Wrought Washer Manufacturing Co., 2110 S. Bay St., Milwaukee, Wis., 60 pages, 4x9 in. Incorporating the results of 50 years of accumulated engineering and mechanical development into one compact manual, this new handbook "Special Washers and Stampings" lists the various shapes, sizes and specifications of more than 19,000 washers, stampings and miscellaneous parts, such as used in the design and manufacture of many types of mechanical products. It is claimed that by the use of the information given in this handbook, the engineer can effect a considerable saving by making his design conform to the specialized units and parts as shown in the pages of this book. Sent free to engineers and executives, upon request.

BOOKS AND BULLETINS

Marketing Industrial Equipment

Bernard Lester. 307 pages, 6x9 in. Bound in cloth boards. Illustrated. Published by McGraw-Hill Book Co., Inc., 330 W. 42d St., New York, N. Y. Price $3.50

Based on a course of lectures given at the graduate school of the University of Pittsburgh, this book presents clearly and concisely the elements of the problem of marketing capital goods as distinct from the quite different problem of distributing consumer goods. The author's long experience in the industrial sales organization of the Westinghouse Electric & Manufacturing Company gives him the background for an authoritative presentation of his subject. He has not hesitated to draw on good sources of facts such as the Marketing Counsellors staff of the McGraw-Hill Publishing Company and *American Machinist*. From the former he has taken the industry key sheet for breaking down the industrial market, and from the latter the results of the 1925 and 1930 surveys of obsolete equipment in the metal-working industry. Mr. Lester's book is so arranged that it can be used as a textbook, but it is primarily a reference book, in our opinion, and as such should be useful to anyone who sells to the industrial or capital goods market.

Tool Steels

James P. Gill. 136 pages, 6x9 in. Illustrated. Published by American Society for Metals, 7016 Euclid Ave., Cleveland, Ohio. Red clothboard covers, price $2.50; paper covered, price $2.

A series of five lectures presented to the National Metals Congress, October, 1934, was primarily intended for the engineer to whom steel problems are highly important. Metallurgical terminology has been avoided to be understandable to laymen, not having expert knowledge of tool steels.

Beginning with a short history of tool steel manufacture in which various methods for melting are portrayed, the opening lecture also contains explanations of casting, forging and rolling, annealing, inspection, classification and characteristics of 14 types of tool steels. Testing is covered in the second lecture, describing a dozen various tests of tool steels, also the effects of alloying elements. The third lecture tells of carbon in low alloy tool steels, while the fourth takes in steels of intermediate alloy content, such as tungsten, chromium and vanadium. In the last lecture, high-carbon, high-chromium and other high-speed steels containing nickel, molybdenum and cobalt are discussed.

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Properties of Tin

International Tin Research and Development Council, *Technical Publication*, Series B, No. 1, by E. S. Hedges and C. E. Homer, 45 pages.

Collected from original literature, data given include atomic, mechanical, thermal, electrical, magnetic and optical properties. Material is presented for the most part in tables accompanied by brief explanatory notes.

Digest of Steels for High Temperature Service

The Timken Steel & Tube Co., Canton, Ohio. Bulletin, 68 pages, 8½x11 in.

A revised and enlarged edition of the original "Timken Digest" published in 1932. All steels in general use for high temperature service, as well as others which seem of interest for the future, have been included in this edition. The data presented are the result of a seven-year program of research in the Department of Engineering Research at the University of Michigan. It contains all essential data of design based on creep strength for a particular rate of creep, on yield strength or proportional limit values as determined by the short-time tensile tests and on temper-brittleness and stability of steels. Chemical composition and physical properties are given, together with 16 graphs. Three articles are included relative to qualifications of steel and factors influencing creep of steel at high temperatures. A chart of SAE alloy steels is also included.

Lubricating Oil Tests

Virginia Polytechnic Institute, *Bulletin*, September, 1934, by J. I. Clower. 41 pages.

An interpretation of the significance of chemical, physical and mechanical tests are outlined, including descriptions of test equipment. Viscosity being the most important property, its measurement is given the most attention. Three mechanical tests are described, and government specifications are reproduced.

Gear Design Calculations–II

· *Bevel Gear Teeth*

CHARLES S. DeLATE

Given the load, the pressure angle, number of teeth in gear and pinion, and the fiber stress, the accompanying example illustrates a method of procedure for calculating the design. Numerical values have been substituted for the various symbols.

Problem—Assume the tractive force is 2,750 lb. at the periphery of a 32-in. wheel. Design a bevel gear and pinion with 14.5 deg. pressure angle, 17 teeth in the pinion, 70 teeth in the gear, to lock the wheel. The fiber stress in the pinion tooth is not to exceed 40,000 lb. per sq.in. when the load is concentrated at the tip of the tooth. Compute:

A—Pitch diameters
B—Thrust loads
C—Radial loads
D—Number of teeth in contact

A—To compute pitch diameters of gear and pinion:

$$W = STyp\left[1 - \frac{T}{m} + \frac{1}{3}\left(\frac{T}{m}\right)^2\right]$$

W = Transmitted tooth load in lb.
S = Fiber stress. lb. per sq. in.
T = Face width of tooth = 2.5 times circular pitch
y = Lewis form factor = 0.20
p = Circular pitch = $\pi/D.P.$
$D.P$ = Diametral pitch (fractional if necessary)
m = Apex distance (slant height of pitch cone)
$2/D.P.$ in. = Working depth
$2.188/D.P.$ = Total depth
$0.560/D.P.$ in. = Gear addendum
$1.241/D.P.$ = Circular thickness of tooth at pitch line

$$m = \sqrt{R^2 + r^2} = \sqrt{\frac{35^2}{D.P.^2} + \frac{8.5^2}{D.P.^2}} = \frac{36}{D.P.}$$

R = Pitch radius for gear = $0.5\,N/D.P.$
r = Pitch radius for pinion = $0.5\,n/D.P.$
N = Number of teeth in gear
n = Number of teeth in pinion

$$W = \frac{\text{Tractive force} \times \text{radius of driving wheel}}{R}$$

$$W = \frac{2{,}750 \times 16 \times D.P.}{35} = 1.257\,D.P.$$

$$\frac{T}{m} = \frac{2.5\pi}{D.P.} \times \frac{D.P.}{36} = 0.2181$$

$$1.257\,D.P. = 40{,}000 \times \frac{2.5\pi}{D.P.} \times 0.2\,\frac{\pi}{D.P.}\,|1 - 0.2181 - 0.0158|$$

$D.P.^3 = 125.7$ $\qquad D.P. = 5.01$

Therefore pitch diameters are as follows:
Gear: $N/D.P. = 70/5.01 = 13.972$ in.
Pinion: $n/D.P. = 17/5.01 = 3.393$ in.

B—Computation of thrust loads on gear and pinion:

$$T_g = W_r\frac{R}{m} \qquad T_p = W_r\frac{r}{m}$$

T_g and T_p = thrust load on the gear and pinion respectively
$W_r = W \times \tan\phi$
ϕ = Pressure angle

$$W = \frac{\text{Tractive force} \times \text{radius of driving wheel}}{R_o}$$

R_o = Distance from the apex to the center of the tooth width measured along the apex distance line
r_o = The same for the pinion as R_o for gear

$$R_o = r_o\frac{N}{n}$$

$$r_o = r\,\frac{1 - \frac{T}{m} + \frac{1}{3}\left(\frac{T}{m}\right)^2}{1 - \frac{1}{2}\left(\frac{T}{m}\right)}$$

$$r_o = \frac{8.5}{D.P.} \times \frac{1 - 0.2181 \times 0.0158}{1 - 0.1091} = \frac{7.6}{D.P.}$$

$$R_o = \frac{7.6}{D.P.} \times \frac{70}{17} = \frac{31.3}{D.P.}$$

$$W = \frac{2{,}750 \times 16 \times D.P.}{31.3} = 1{,}405\,D.P.$$

$W = 1{,}405 \times 5.01 = 7{,}040$ lb.
$W_r = 7{,}040 \times \tan 14.5$ deg. $= 1{,}820$ lb.

$$T_g = 1{,}820 \times \frac{35}{D.P.} \times \frac{D.P.}{36} = 1{,}770 \text{ lb.}$$

$$T_p = 1{,}820 \times \frac{8.5}{D.P.} \times \frac{D.P.}{36} = 430 \text{ lb.}$$

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C—Computation of radial loads on gear and pinion:

$R_g = \sqrt{W^2 + T_p^2}$

$= \sqrt{7{,}040^2 + 430^2} = 7{,}050$ lb.

$R_p = \sqrt{W^2 + T_g^2}$

$= \sqrt{7{,}040^2 + 1{,}770^2} = 7{,}250$ lb.

D—Check for interference. If the left-hand side of the equation is less than the right-hand side, there is no interference:

$$2R'A + A^2 < (2R' + r')\, r' \sin^2 \phi$$

r' = Formative pitch radius of pinion
R' = Formative pitch radius of gear
A = Gear addendum

$$r' = \frac{r}{\rho}\sqrt{\rho^2 + 1}$$

$$\rho = \frac{N}{n} = \frac{70}{17} = 4.11 \qquad \rho^2 = 16.9$$

$$r' = \frac{1.6965}{4.11}\sqrt{17.9} = 1.748 \text{ in.}$$

$$R' = R\sqrt{\rho^2 + 1} = 6.986 \times 4.235 = 29.55 \text{ in.}$$

$$A = \frac{0.560}{D.P.} = \frac{0.560}{5.01} = 0.112 \text{ in.}$$

$2 \times 29.55 \times 0.112 + 0.112^2 < (2 \times 29.55 + 1.748)$
1.748×0.25038^2
$6.63755 < 6.64$, (approaches interference)

Angle of approach on true pinion

$$= 57.3 \tan \phi \frac{r'}{r}$$

$$= 57.3 \times 0.2586 \times \frac{1.748}{1.6965} = 15.25 \text{ deg.}$$

57.3 = Number of deg. in one radian

Length of line or arc of action on formative pinion

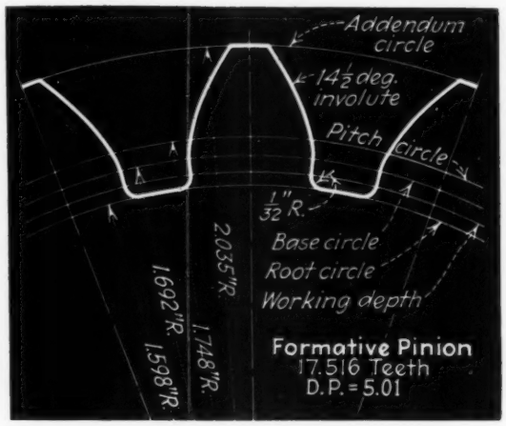
$= \sqrt{(r' + a)^2 - b^2}$

$b = r' \cos \phi$ = base circle radius
a = Pinion addendum
$a = 2/D.P. - A = 2/5.01 - 0.112 = 0.287$ in.
$\sqrt{(1.748 + 0.287)^2 - (1.748 \times 0.96815)^2} = 1.123$ in.

Total angle of contact on true pinion

$$= \frac{\text{Length of line of action} \times 57.3}{r \cos \phi}$$

$$= \frac{1.123 \times 57.3}{1.6965 \times 0.96815} = 39 \text{ deg. } 12 \text{ min.}$$



Angle of recess on true pinion
Angle of contact — angle of approach
39 deg. 12 min. — 15 deg. 15 min. = 23 deg. 57 min.
Number of teeth in contact

$$= \frac{\text{Angle of contact}}{360} \times n = \frac{39.2 \times 17}{360} = 1.850$$

Layout the pinion tooth profile

When laying out a bevel gear tooth profile, the dimensions for the formative tooth are used.

r^1 = pitch radius = 1.748 in.
Addendum circle radius = $r^1 + a$
= 1.748 + 0.287 = 2.035 in.

Base circle radius = $r^1 \cos \phi$

$= 1.748 \times 0.96815 = 1.692$ in.

Root circle radius = addendum circle radius — tooth depth = $2.035 - 2.188/D.P. = 1.598$ in.

Formative number of teeth on pinion

$$= \frac{n}{\rho}\sqrt{\rho^2 + 1} = \frac{17}{4.11}\sqrt{17.9} = 17.516$$

The circular tooth space width of the pinion at the pitch line equals the circular tooth thickness of the gear at the

pitch line $= \frac{1.241}{D.P.} = 0.248$ in.

Angle between tooth center lines
$= 360/17.516 = 20.555$ deg.
Working depth $= 2/D.P. = 0.3992$

The May number of *Product Engineering* will contain examples of the design of straight spur tooth gears made of hardened alloy steel, consideration being given to the wear factor rather than the strength of the tooth.

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